VOLUME 30 NUMBER 5

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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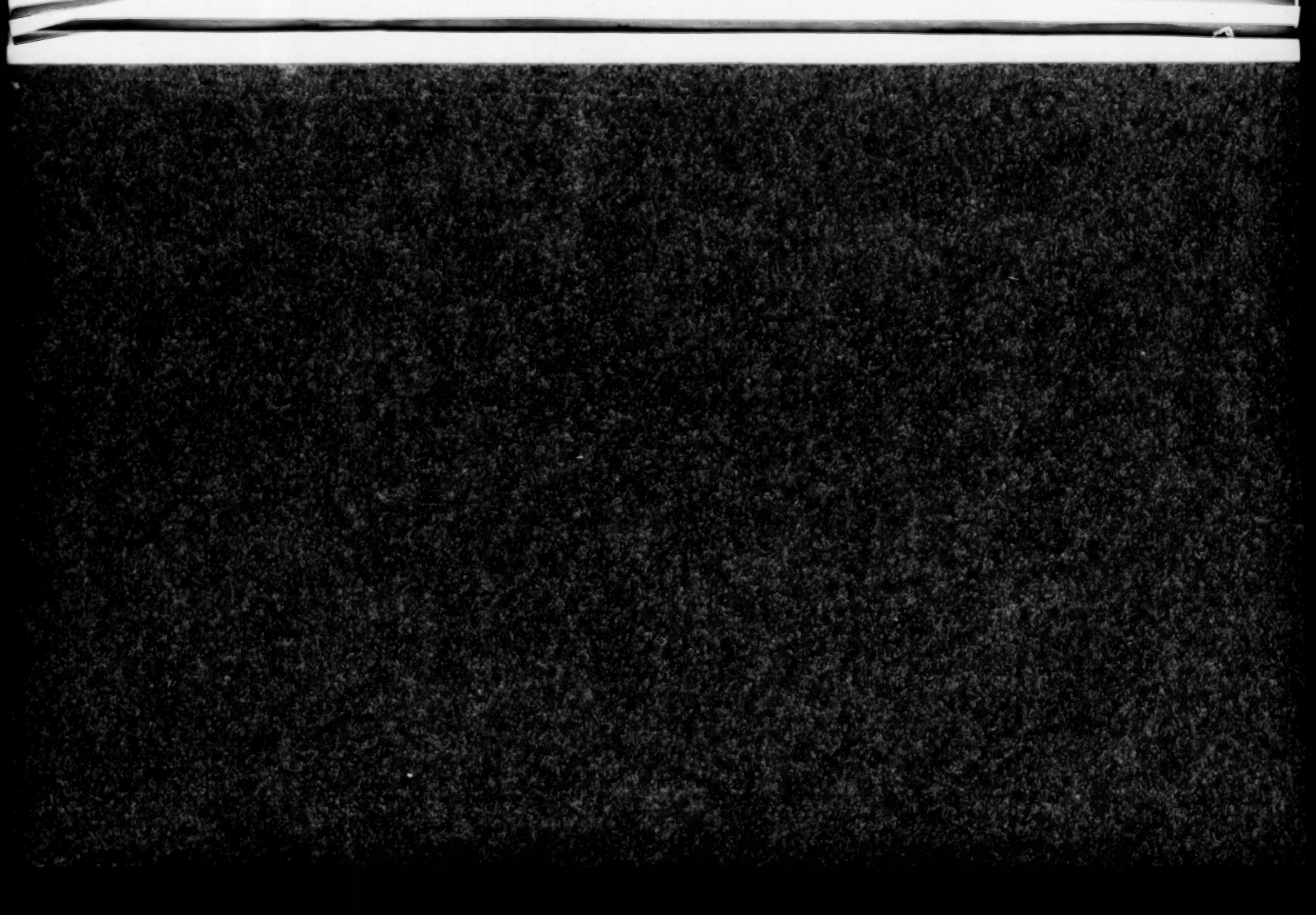

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THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

PROCEEDINGS

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P. W. GATES — W. F. M. GOSS
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PROCEEDINGS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 30 MAY 1908 NUMBER 5

THE next monthly meeting of the Society will be held in the Engineering Societies Building, Tuesday evening, May 12. A paper upon "Clutches," with special reference to the types used on automobiles, will be read by Henry Souther, of Hartford, Conn. Lantern slides will be used to show the development of the various types. The text of the paper appears in this number of Proceedings.

The meeting will be of importance not only to those directly interested in automobile construction, but to all who have to do with the use of clutches for any other purpose, such as in machine tool work, power transmission, hoisting machinery, textile and other classes of machinery.

The requirements for a successful clutch are so much more difficult to meet in an automobile than in almost any other type of machine, that clutch design has become an engineering problem since the advent of the automobile, meriting the attention of the engineering profession.

The meeting will afford an opportunity for a full discussion of the design and use of clutches of various types for different kinds of machinery and it is earnestly desired that those who have had experience with clutches will attend and contribute the results of their experience for the benefit of the membership. The discussion upon clutches will be continued at the June meeting in Detroit.

REGULAR MEETING OF THE COUNCIL

April 14, 1908, Mr. Fred J. Miller, Vice President, presiding; and present also Messrs. Taylor, Hutton, Wiley, Riker and Rice.

Resignation of E. H. Powell accepted.

Addition proposed to By-Law 18 as follows:

The Secretary shall set aside $5 per year out of the dues of each member as a subscription to Proceedings.

The necessity for this is the requirement of the Post Office Department that this formality be complied with in order that the Society shall enjoy the full privileges of second class mail matter for Proceedings, soon to be called The Journal of The American Society of Mechanical Engineers.

Voted, That professional records be approved as prepared by the Membership Committee, subject to any subsequent communications, which shall be considered authority to refer back to the committee.

Voted, That the Society resign from membership in other bodies.

Voted, To publish in Proceedings all cases of death, resignation or other severance of relations with the Society.

Voted, That the Committee on Affiliated Societies be appointed as a special conference committee with relation to the Society of Automobile Engineers.

Voted, To appoint Mr. W. E. Smith, member of the Society at St. Petersburg, Honorary Vice President to represent the Society at the Eleventh International Congress of Navigation to be held at St. Petersburg, May 31 to June 7, 1908.

Research Committee:

Voted, To suggest to the President that he extend an invitation to the membership to suggest nominations for a Research Committee.

Other Societies: The exchange of courtesies with other societies was recommended and the Secretary invited to secure such exchange wherever possible and make it a feature of Society work.

Advertising Committee: The unanimous recommendation was received of Mr. E. J. Gibling as advertising manager, and approved.

Communications: From the Chamber of Commerçe, Pittsburg, a copy of resolutions passed by them on the subject of conservation of natural resources. The Secretary responded by sending a copy of the Society's resolutions and the program of the meeting of April 14.

Resolved, That as The Pratt & Whitney Company, Hartford, Conn., have presented to the Society a set of standard machine screw gages, plugs and templets, they be accepted and the thanks of the Council be transmitted to The Pratt & Whitney Company.

Adjourned.

CONSERVATION MEETINGS

The general interest among engineers on the subject of the conservation of the natural resources of the United States is indicated by the attention this subject is receiving at the meetings of engineers' associations. Not only has it been discussed at the New York meetings of the American Institute of Electrical Engineers and The American Society of Mechanical Engineers, but on April 15 the Engineers' Club of St. Louis held a meeting on conservation. At this meeting President Holman of The American Society of Mechanical Engineers opened and closed the discussion.

THE SPRING MEETING AT DETROIT

To be Held June 23–26

As previously stated the Spring Meeting at Detroit will open on the evening of June 23 and end Friday afternoon, June 26. Simultaneous with this convention will be the meetings of the Society for the Promotion of Engineering Education, of the Society of Automobile Engineers and of the mechanical branch of the Association of Licensed Automobile Manufacturers. The meetings will be arranged, as far as possible so that members interested in subjects presented by the other Societies may attend their sessions without missing papers on related subjects read before their own Society.

One session will be devoted to a symposium on the conveying of materials when papers will be presented upon hoisting and conveying machinery including belt conveyers, the use of conveying machinery in cement plants, etc.

Papers will also be presented on the following subjects:

"Clutches, with Special Reference to Automobile Clutches," by Henry Souther.

"Some Pitot Tube Studies," by W. B. Gregory and E. W. Schoder.

"Thermal Properties of Superheated Steam," by R. C. H. Heck.

"Horse Power, Friction Losses and Efficiencies of Gas and Oil Engines," by L. S. Marks.

"A Journal Friction Measuring Machine," by Henry Hess.

"A Simple Method of Cleaning Gas Conduits," by W. D. Mount.

"A Rational Method of Checking Conical Pistons," by G. H. Shepard.

"The By-Product Coke Oven," by W. H. Blauvelt.

Among other subjects upon which papers are expected are gas engines, steam engine and turbine tests, and condensers.

The convention will open with an informal reception and a brief business session on Tuesday evening. On Wednesday evening there will be a lecture by Prof. John A. Brashear, manufacturer of astronomical and physical instruments, Allegheny, Pa., who will take as his

subject, "Contributions of Photography to our Knowledge of Stellar Evolution." On Thursday evening will be the main reception.

Opportunity will be offered for inspecting the interesting manufacturing plants of the city, including the automobile factories, the factory of the Burroughs Adding Machine Company, the shipbuilding yards, and the several establishments manufacturing medical preparations.

Detroit abounds in opportunities for outdoor excursions on both land and water and the local committees will provide abundant opportunity for the members and their guests to visit the attractive places in the city and vicinity during the intervals of the convention. Among the excursions planned is one to the University of Michigan at Ann Arbor, where is located the large naval testing tank.

RAILWAY TRANSPORTATION NOTICE

Arrangements for hotel, transportation and sleeping car accommodations should be made personally by each person.

SPECIAL TRAIN ARRANGEMENTS

One or more private Pullman sleeping cars will be reserved on each of three trains from the East. These reservations can be made however only in case the number of applications received for accommodations reach the minimum set by the Pullman Company. Address applications to Mr. S. E. Whitaker, Office Manager.

The first train selected is the Wolverine via N. Y. C. R. R., with buffet, smoking and library car, and dining car. The Pullman sleeping car will run through to Detroit without change. This train will leave New York on Monday and arrive in Detroit Tuesday morning. The first session of the convention will be on Tuesday evening.

SCHEDULE

Lv. New York, Monday, June 22	4.30 p. m.
Lv. Boston	1.45 p. m.
Lv. Albany	8.00 p. m.
Lv. Syracuse	11.40 p. m.
Lv. Schenectady	8.31 p. m.
Ar. Detroit, Tuesday June 23	8.15 a. m.

The second train selected is the Chicago Limited via Pennsylvania Railroad, with smoking and dining cars. The Pullman sleeping car will run through to Detroit if 18 reservations are made.

Lv. New York, Monday, June 22	4.55 p. m.
Lv. North Philadelphia	7.08 p. m.
Lv. Washington	5.45 p. m.
Lv. Baltimore	7.05 p. m.
Ar. Detroit, Tuesday June 23	4.10 p. m.

The third train selected is the Chicago Toronto Express via Lehigh Valley Railroad. This line allows stop-over at Niagara Falls on individual tickets.

Lv. New York, Monday, June 22	5.40 p. m.
Lv. Philadelphia	6.30 p. m.
Lv. Bethlehem	8.33 p. m.
Lv. Buffalo, Tuesday, June 23	5.35 a. m.
Lv. Niagara Falls	7.10 a. m.
Ar. Detroit, Tuesday June 23	1.20 p. m.

An alternative train is the Buffalo and Cleveland special, via New York Central Railroad, leaving New York at 8 p.m. Monday, and arriving in Detroit at 1.30 p.m., Tuesday.

REGARDING SPECIAL RATES

Application for special reduced rates has been made to the various Passenger Associations. Detailed announcement of these rates will appear in the Bulletin of the Spring Meeting, which will be mailed to each member in May.

HOTEL ACCOMMODATIONS

The Cadillac has been selected as the headquarters of the Society during the convention and also as the meeting place for all the professional sessions.

The rates at the Cadillac during the convention will be:

American plan, room without bath, $3.50, $4 to $4.50, each person. Room with bath, $4, $4.50, $5 to $8, each person. European plan, $2 and upward per day, each person. A special rate of $3 per day each person, American plan, and $2 per day, each person, European plan, will be made where two or more persons occupy a room without bath.

The rates of the Hotel Pontchartrain will be:

European plan, room without bath $2 and $2.50 per day for one person; for two persons $3 and $4 per day. Room with bath $3, $3.50, $4 and $5 per day for one person; for two persons $5, $6, $7 and $8 per day.

Another hotel recommended is the Tuller, located at Adams Avenue and Park Street, about five minutes' walk from headquarters. The rates will be:

European plan, $2 per day for one person occupying a room alone; $1.50 per day for each person when two occupy a room. Every room is connected with bath.

The rates of the Hotel Normandie, located Congress Street and Woodward avenue, within five minutes' walk from headquarters, will be:

American plan, rooms without bath, single $2.50 to $3 per day, double $4.50 to $5.50 per day; rooms with bath, single $3 to $4 per day, double $5 to $7 per day. European plan, rooms without bath, single $1 to $2 per day; double $1.50 to $3 per day; rooms with bath single $2 to $2.50 per day, double $3 to $4 per day.

Members expecting to attend the meeting and desiring accommodations at the above hotels will please communicate directly with the hotel. Members are advised to write immediately, thus avoiding disappointment as to the accommodations they may desire.

APRIL MEETING ON THE CONSERVATION OF OUR NATURAL RESOURCES

The April meeting of the Society on The Conservation of Our Natural Resources was held, as previously announced, in response to the invitation of the President of the United States to coöperate for securing the conservation of the natural resources of our country.

The Society extended a general invitation to members of the engineering profession to attend this meeting and prominent representatives of the four national engineering societies were seated on the platform with the speakers of the evening.

Four addresses were given as follows:

"The Conservation of the Waters and Woods," by Dr. W J McGee, Secretary of the Inland Waterways Commission.

"The Conservation of the Nation's Fuel Supply," by Dr. W. F. M. Goss, Dean of the College of Engineering, University of Illinois.

"The Conservation of Stream Flow, Water Power, and Navigation," by Dr. George F. Swain, Director of the Department of Civil Engineering, Massachusetts Institute of Technology.

"The Relation of the Engineer to the Body Politic," by Dr. Henry S. Pritchett, president of the Carnegie Foundation.

At the close of the addresses a number of lantern slides was shown by Dr. McGee.

Mr. John W. Lieb, Jr., Vice-President of the Society, was chairman of the meeting and in his opening remarks extended a welcome to the distinguished speakers, to the officers and representatives of sister engineering societies and to the guests.

Several letters were read by the Secretary, among which was the following from the President of the United States:

THE WHITE HOUSE, WASHINGTON

April 13, 1908.

MR. CALVIN W. RICE, *Secretary, The American Society of Mechanical Engineers, 29 West 39th Street, New York.*

My dear Mr. Rice:

I regret that time did not permit me when you were here to tell you how gratified I am with the zeal and interest which the members of the engineering profession are taking in the movement to conserve our natural resources.

It is the duty of all our citizens to exercise foresight in these matters, but it is peculiarly the duty of the engineer, who is in fact a trustee of the forces of nature. Above all men the engineer is responsible for seeing that our natural resources are not wasted and that their development is for the common good and for the continuance of the welfare of our citizens.

I was particularly pleased with the resolutions passed by your Council and indeed with the coöperation of all engineering bodies in bringing this matter forcibly to the attention of all the people. I am sorry that I can not be with you on Tuesday evening to hear the addresses, but I extend to all the engineers my heartiest good wishes.

Sincerely yours,

[Signed] THEODORE ROOSEVELT

The following telegram was received from President Schurman of Cornell University, recently appointed by the Governor of New York as one of the delegates to accompany him to the Washington conference:

ITHACA, N. Y., April 14, 1908.

SECRETARY CALVIN W. RICE, *39 West 39th Street, New York.*

Regret I cannot attend engineers meeting today. I attach greatest importance to your deliberations on conservation of the country's natural resources.

J. G. SCHURMAN.

Mr. Lieb then announced the purpose of the meeting to discuss the nature and extent of our resources; the demands which have been made upon them in the past, and which may be expected in the future; whether our use of them has been economical or wasteful; and suggestions in the direction of future public policies which may lead to a more efficient utilization and more wise conservation of our natural resources. He reminded the audience that our first President, the immortal Washington, himself an engineer of ability, was at the time of his election President of the Delaware and Chesapeake Canal Company, and now the President of the United States following in the footsteps of his predecessor has created a commission on inland waterways whose duty it is to formulate a broad policy for the protection and control of our water supply.

Then followed the first address, by Dr. W J McGee, Secretary of the Inland Waterways Commission.

THE CONSERVATION OF THE WATERS AND WOODS, BY DR. W J MCGEE

Dr. McGee pronounced the subject of conservation, touching, as it does, the conditions of the perpetuity of our Nation, the most important we have ever had to consider. He said what the country

specially needs today is a mental and moral revolution and awakening by the individual to a sense of obligation for the preservation of the gifts of nature, not only for the present generation, but for all the generations to come. He spoke especially of the water as one of the important resources to be conserved. The average rainfall over the mainland of the United States is about thirty inches, or about 200 000-000 000 000 cu. ft. per year, which would make about ten lower Mississippis. Just as soon as our population and industries have increased to such an extent as to consume this annual rainfall, we shall have reached the limit of our development, unless by that time human ingenuity has devised some way, recognized by only a few of our scientists at the present time, of producing water in addition to that we receive from the heavens.

Dr. McGee said that about three-fifths of the rainfall is evaporated and serves to temper the air, producing dews, fogs, and regulating the temperature; about one-fifth passes deeply into the earth, or is consumed in various chemical combinations on the earth's surface; the other fifth flows down to the sea. It is for the conservation of this one-fifth that the people of the United States should be most concerned.

There is no diminution in the rainfall or the precipitation on the surface, but too much water flows away on the surface, producing torrents in the streams and the chain of evils following.

Our injudicious treatment of water, our failure to appreciate its importance is greatly influencing navigation. The Ohio river is an example. At Cincinnati it has a range from low water to flood water of more than 50 ft., so that it is extremely difficult to maintain terminals in such a manner as to render the Ohio an effectively navigable stream. These floods are constantly increasing because of the deforestation at the headwaters of mountains and foothills in which the waters first gather, so that the soil is no longer a great sponge, drinking in the water and giving it back slowly, but sheds it with a rapidity which gathers it in torrents, and the torrents in turn take millions of tons of soil on the way and deposit them in the channels of the stream, an obstruction to navigation. The navigability of our streams on the whole today, is much less than it was at the time of the first settlement of each district.

Dr. McGee further said: The soil is one of the assets of the earth which is only lately recognized by this nation. It is the original vegetable humus, the accumulation of centuries, even of milleniums. The Mississippi Valley alone is losing every year more than 500 000-

000 tons of richest soil matter through soil erosion, and the entire United States is losing somewhere between 1 000 000 000 and 2 000 000 000 tons. Considering the lowest price of this soil as a fertilizer, which cannot be less than $1 a ton, the farmers of the United States, through soil erosion alone, are paying an annual tax of between $1 000 000 000 and $2 000 000 000, which is absolutely without return.

The loss of our soil means the loss of our forests; the exhaustion of our iron ores; the exhaustion of our coals. To consider one relation between coal, iron and water, where rivers are navigable, it requires only about one-tenth as much iron to carry a given cargo of freight by water as to carry the same cargo by rail, so that improvement in our navigation will diminish the tax on our iron ores. It requires to carry each ton of freight only about one-eighth (depending of course on the speed movement), as much coal to carry the freight on water, as is required to carry the same freight by rail, so that improvement of navigation will cut down the rate of consumption of coal, at least relatively. These are among the relations that might be multiplied indefinitely.

THE CONSERVATION OF THE NATION'S FUEL SUPPLY, BY DR. W. F. M. GOSS

This subject was presented by the speaker under four headings: The Value of Fuels, The Production of Fuels, Lack of Economy in the Use of Fuel, and How Economy of Fuel is to be Secured.

Under the first heading striking illustrations were given of the magnitude of the mining and allied industries. In 1850 there were mined in the United States 6 000 000 tons of coal, and since this date the annual production has more than doubled every ten years, until in 1906 it reached the enormous amount of 414 000 000 tons.

In addition to these 414 000 000 tons of anthracite and bituminous coal mined, there were drawn to the surface nearly 400 000 000 000 cu. ft. of natural gas weighing over 9 000 000 tons and 126 000 000 barrels of oil weighing approximately 17 000 000 tons. Summarizing these gives the enormous total of 440 000 000 tons, an amount so great as to challenge the imagination.

Fancy it all, for example, to have the form of marketable size bituminous coal and to be dumped into a windrow of triangular cross section, piled to a height of 32 ft, with a width of base of 46 ft. Such a pile would be as high as an ordinary two-story pitched roof house.

It would contain nearly 30 tons per foot run, while the length of the windrow would be sufficient to reach from New York to San Francisco. This measures the annual rate of production at the present day only; what will be the rate 10, 50, or 100 years hence?

Under the second heading Dr. Goss recounted some of the wasteful methods of mining. The tendency in many districts is to accomplish with powder what ought to be done with pick or machine. Excessive charges are common, which increases the percentage of fine coal, in many mines never hoisted. Heavy explosions not infrequently bring down the roof of the mine over large areas of coal which under present day conditions cannot be successfully opened up again. When a good layer of coal is overlaid by one or more thinner layers which cannot be worked with the same facility as the layer below, it is common practice to take out the coal below and to permit the roof above to cave in, destroying the continuity of the thinner layers above, making subsequent mining operations impossible.

Or the coal above may be heavy and of satisfactory quality, but it may be seamed with thin layers of shale or slate, and the expense of separating it from its impurities makes it unattractive and it is consequently not taken out. Again, the presence of abnormal amounts of sulphur in portions of a layer of coal otherwise satisfactory may result in the partial removal of a single heavy layer, the undesirable portions being left as abandoned property. It is estimated by one high in authority,[1] whose experience has been chiefly in the Appalachian field, that approximately 50 per cent of the coal in the mines never gets to the surface.

In the production of oil actual waste has probably not been great, although the supply has been heavily drawn upon. The waste of natural gas, however, has been very great. In 1891 it was estimated by Mr. Gorby, State Geologist of Indiana that the amount of gas daily consumed at the wells and actually wasted was in excess of 100 000 000 cu. ft. Since that time the wastes have been allowed to continue. Twelve years later a pipe line delivering 1 000 000 cu. ft, a day was found to be leaking six times the volume it delivered. As the supply of gas diminished, oil appeared and old gas wells were uncapped for the purpose of blowing down the pressure to promote the flow of oil. Under the influence of heavy legitimate consumption and of these wastes, the Indiana field is now, seventeen years after Mr. Gorby made the statement above referred to, practically

[1] I. C. White, State Geologist of West Virginia.

dry. Fuel sufficient in quantity to have served generations of people was squandered in a period of two decades.

Under the third topic the speaker called attention to the fact that in every movement of fuel, from the point of production to the stack of the furnace in which it is used, loss occurs which, by the exercise of attention, may be reduced or entirely eliminated.

The greater part of the fuel supply is consumed in industrial processes, a comparatively small quantity being required to keep the nation warm. Nearly one-fourth of the total output, or about 100 000 000 tons, goes to the railroads and is for the most part consumed in locomotive fireboxes; that is, 51 000 locomotives are today burning coal at a rate which is in excess of the rate of total production 25 years ago. Approximately 20 per cent of the coal supplied locomotives is used in starting fires, in keeping the machine hot while standing on side tracks, or is left in the firebox at the end of the run. Sixteen million tons of the annual consumption are thus accounted for. From 8 to 10 per cent of the remainder is discharged as unconsumed fuel from the stack during the operation of the locomotive and the remainder is required for the generation of steam.

If American locomotives were of a more highly developed type such as are much employed for foreign service, if they were designed with compound cylinders or with superheaters, the amount of fuel required would be less and the annual coal consumption would be reduced 6 000 000 to 10 000 000 tons.

The use of coal in the manufacturing industries of the country is on the whole probably not attended by a higher degree of efficiency than that which prevails on railways. Everywhere there are evidences of bad or imperfect practices.

In pointing out how economy of coal is to be secured, Dr. Goss said the necessary steps are:

a Scientific research for the establishment of facts;

b Practical development of the facts thus developed on a scale which will convince men that there is profit, direct or indirect, in a better practice;

c Restrictive legislation which will protect the public from the competition of unscrupulous men;

d Effective inspection which will secure enforcement of laws.

The process must begin with education—not with coercion.

The speaker then showed how improvement could be secured at the mines through a better understanding of the use of explosives and regulation of their use, referring to the improved practice in

England in this regard. Processes must be developed for working coal above ground whereby inferior grades may be better adapted to use. Fine coal must be washed and if this is not sufficient to give them a market they must be coked or briquetted. Slaty and sulphurous coals must be crushed and washed and processes now undiscovered must be called into service to make it profitable to hoist from the mine every ounce of available fuel. Wherever it can be shown that such processes are practicable, it is but reasonable to compel their use.

In the matter of the economical use of fuel we still have much to learn, a statement emphasized by the fact that the problem relates largely to the use of bituminous fuels, of all fuels the most difficult to handle. The task of learning how to burn efficiently and without smoke the good and bad soft coals of our country is a task assigned to the present generation, and The American Society of Mechanical Engineers constitutes the strongest single influence in this work.

It has not yet been shown as a commercial proposition how small, soft coal fires under boilers can be maintained without smoke. No laws have been deduced to define the desirable length of flameway for coals of different compositions. The possibility of a much more general application of the gas producer, both as an individual piece of apparatus and as a detail in furnace construction, are yet before us. These and many similar matters are fundamental in the development of a more economical practice.

The conservation of our fuel resources is an engineering problem and it will yield to treatment in proportion to the engineering skill which is concentrated upon it. The matter is of national importance and should receive national attention.

THE CONSERVATION OF STREAM FLOW, WATER POWER AND NAVIGATION, BY PROF. GEORGE F. SWAIN

In opening his address Professor Swain drew a parallel between the individual and the nation, and said it is always a delicate task to induce either an individual or a people to live prudently and husband its resources. This country, like almost all others, has erred both in unduly drawing upon its natural resources beyond the power of reproduction to support the consumption, when such power exists; and also by waste and extravagance, unaccompanied by any corresponding good.

Turning to the subject of water, he said that when the country had

a small population pure water could be had in unlimited quantity. But with the recklessness characteristic of the human race we allowed our streams to become polluted, and the public health endangered. Our streams should not only be preserved in volume, but in purity. Much depends, also, upon the preservation of the regularity of stream flow, and streams may be preserved and regulated in two ways:

a By the preservation of forests;

b By constructing reservoirs.

The preservation of forests appeals to us both as a source of supply of important material and as a means of affecting stream flow. Waste and fire have so reduced our originally magnificent forests that now we are told there is but 20 years' supply left.

It is improbable that forests increase the total rainfall or run-off. Indeed there are reasons for thinking that they sometimes decrease the total run-off. Trees intercept about one-fourth the total rainfall by their leaves and branches, not allowing it to reach the ground; they also absorb considerable moisture by their roots, to be evaporated through their leaves; but these losses are more than compensated for by the reduction of evaporation of water from the ground or from water surfaces, which is only about one-fourth to one-third as great in forests as in cleared ground. The forests act as a shield against the wind, cool the air and increase the relative humidity. Some writers claim that forests evaporate immense quantities of water, but as Ebermayer found the total amount of moisture in the air to be almost the same in forests as in cleared ground, we may dismiss this contention; and we furthermore have the conclusion of Risler that forests evaporate less than grass land.

Facts were also cited to prove that forests are regulators of flow and that whether they increase or decrease the total run-off is unimportant; for the value of a stream depends not upon its total flow, but upon the regularity of that flow. The humus of a forest is a great sponge which absorbs the water and gradually gives it out.

Examples were quoted of the effect of removing forests. In France a small spring flowed from a cleared area, a forest of firs was allowed to grow up, the spring broke forth in much larger volume, and for 40 or 50 years was considered the best in the neighborhood. Finally the woods were cut off, the spring dwindled, and conditions became as they were 90 years before. Becquerel, Marsh, and other writers give many examples showing that the cutting down of the forests has been followed by the drying up of springs, the lowering of lakes, and an increasing irregularity in the flow of rivers.

Hough, in his Elements of Forestry says:

> The Khanate of Bucharia presents a striking example of the consequences brought upon a country by clearings. Within a period of 30 years this was one of the most fertile regions of central Asia, a country which, when well wooded and watered, was a terrestrial paradise. But within the last 25 years a mania for clearing has seized upon its inhabitants and all the great forests have been cut away, and the little that remained was ravaged by fire during a civil war. The consequence was not long in following and has transformed the country into a kind of arid desert. The water courses are dried up and the irrigating canals empty. The moving sands of the desert, being no longer restrained by barriers of forests, are every day gaining upon the land, and will finish by transforming it into a desert as desolate as the solitudes that separate it from Khiva.

Forests diminish the violence of floods. A wooded surface can never pour forth such deluges as flow from cleared land, or especially from unwooded mountain slopes, and the forests on these should be protected, especially as such land is not suited to agriculture.

> In the French Department of Lozere, which was among those most severely injured by the inundations of 1866, it was everywhere remarked that the ground covered with wood sustained no damage, even on the steepest slopes, while in cleared and cultivated fields the very soil was washed away and the rocks laid bare by the pouring rain.

If forests are cut down and not replaced, we must therefore expect an increasing irregularity of flow of streams. We must not, however, claim too much, or expect that, by allowing forests to grow freshets can be entirely done away with. If fire follows lumbering, as so often the case, the soil is damaged and the trees cannot grow upon it for a long period.

It is not necessary to emphasize the importance of water power to an audience of engineers. Many of our industries came into existence because of it and the cities of Lawrence, Lowell and Manchester would very likely never have been founded in their present locations but for the existence there of great waterfalls.

In the five streams with their tributaries which drain the White Mountain region, it is estimated that 350 000 h.p. is utilized, this being about 20 per cent of the total utilized water power of the country, and nearly 3 per cent of all the utilized power of the country. In the Southern States the development has been also great, especially in recent years, but there are still greater possibilities. The United States Geological Survey estimates that there is a minimum of 2 800-000 h.p. generated on streams draining the Southern Appalachians, of which one-half or 1 400 000 h.p., can be economically developed.

At $20 per horse power per annum this represents $28 000 000 per annum. This estimate is given simply to indicate that even relatively small injuries to water powers may result in a great injury in the aggregate.

The second means of controlling stream flow is by storage reservoirs. This method has not yet been undertaken to any large extent, though every mill pond is a storage basin. This method, however, will naturally be used more and more and the reservoirs will be located near the headwaters of the streams where the floods have their origin. Here the effect of the destruction of forests is most injurious, for from the steep unwooded slopes the rain and melting snows will carry off great quantities of earth and deposit it in the rivers and reservoirs; and the best part of the earth, fertile top soil, will be carried off in this way, not only filling up the streams, but making the soil incapable for a long time of supporting a forest growth. This silting up is especially to be feared in the Southern Appalachians, where the soil is deep to the mountain tops, but it is also of importance in the White Mountains. We spend great sums for dredging the lower reaches of our rivers, whereas the preservation of our forests would render unnecessary some of this work at least.

What should be our position as engineers on matters of this kind? Should we not exert ourselves to do our full share both in the development and conservation of our natural resources? Engineering is a noble profession. Engineers as a rule are honest, intelligent, capable men. They have been trained to cope with the forces of nature and to direct them for the use and convenience of man. They have been taught to seek the truth and not to make the worse seem the better reason. This is the age of the engineer, but the engineer has not yet come to his own. He is too often looked upon as a servant or employee, a tool for others to use, not a leader. This is perhaps to some extent due to engineers themselves. Too frequently they are so much absorbed in details of their profession that they do not grasp, or seek to grasp, the broader problem of which their work is a part.

In the development and conservation of the nation's resources is it not to our own profession that the nation should look for the men, who by training and knowledge are best fitted to show the way? An individual responsibility rests upon us to take an active, interested enthusiastic part, working on and directing public opinion instead of being subservient to it.

THE RELATION OF THE ENGINEER TO THE BODY POLITIC, BY DR. HENRY S. PRITCHETT

A profession as distinguished from a business implies a vocation in which not only is expert service applied for the benefit of him who uses it, but also in the interest of the state and the public. We are prone to consider the essential difference between the work of the engineer of today and the engineer of the past to lie in the greater achievement which the engineer of the present day may compass through the largeness of his enterprises and the rapidity with which great projects are carried out. This way of thinking is, I think, essentially wrong. I doubt whether the engineer of today deserves any great honor, as compared with the engineer of a thousand years ago, from the mere consideration of the size of his operations. An engineer is a man who applies the science of his time to the construction of works which have to do with the needs of his day. Measured by this standard, the engineer of today can boast very little over the engineers who raised the pyramids, who built the temples of Karnak or the hanging gardens of Babylon, in the mere matter of size.

The great difference between the accomplishments of the engineer of today and of the engineer who built the pyramids lies not in the difference of magnitude, but in the difference of purpose in which their respective tasks were undertaken. The engineer of a thousand years ago wrought as the servant of a king, did the things that the king commanded, and these things in many cases were associated with enterprises of little moment for civilization or for improvement.

The engineer of today works also in employment, not always of a king, but of an employer or a company; but his work differs from that of the engineers of a thousand years ago in the fact that he works always in the service of mankind. The building of the pyramids was a splendid engineering feat, which always excites our admiration, but it had small significance in promoting the comfort, happiness or civilization of the people for whom they were built. There was celebrated in this city last week an event much less picturesque: the opening of the four tunnels which connect the city of New York with Long Island, and yet this latter work of the engineer has a far larger public interest and is of far higher public service.

Engineering has grown from a vocation to a profession. It has entered into the company of those great callings whose members are recognized as not only the servants of those who employ them, but as the guardians, also, of the public interest and the public honor.

There can be no question that professions in our social order rise to great power only by reason by the strict sense of honor of their members and again lose relatively by a lack of observance of that sense. There is no truer word than that of Bacon, "I hold every man a debtor to his profession."

It is for this standing of engineering as a profession, rather than a business, for a sense of honor founded on the highest professional ideals, rather than in the exigencies of business life, which I wish to urge upon the engineers of our day.

And what is the practical significance of such a distinction in this matter which we are considering? Is the engineer to decline to carry out the wishes of his employer? Is he to set himself up as the judge of that which is fitting and right and profitable, for those who pay his salary?

The answer to these questions lies in the use of fair judgment, a sound sense of justice, and a quick appreciation of the larger public causes. The engineer is in a unique position to exercise by his advice, suggestion and his consciousness of the public interest, a great influence in the encouragement of justice and of wisdom. It will be a part of the honor due to his profession to look always at the larger rather than at the smaller view of development; to undertake the consideration of great enterprises rather from the standpoint of the great and unlimited future than from the standpoint of the small and limited present. In a word he will, if he be a true member of a profession, while serving loyally his employer, keep ever before the eye not only of himself, but of those whom he serves, the honor of his own profession, the debt which he owes to it, and the service of the larger interests of humanity which these considerations require.

NECROLOGY

EDWARD BETTS BRISLEY

Edward Betts Brisley was born in New York, July 23, 1880. He attended the Dwight School in New York, preparatory school in Hoboken, New Jersey, and entered the Stevens Institute of Technology in February 1898, graduating in 1902. After leaving college he entered the employ of the Manhattan Railway Company as engineer of erection at the company's yards at 156th Street, New York. In April 1903 he was engaged by the Interborough Rapid Transit Company as Inspector of Construction of the Subway Power House at West 59th Street. A year later he accepted a position with the Crocker-Wheeler Company at Ampere, New Jersey, and afterward was transferred to the company's office in Pittsburg.

In February 1906 he became associated with R. M. Bailey & Co. of Philadelphia. In July of the same year Mr. Brisley became one of the partners who established the Standard Engineering Corporation, with which he was connected until his death.

Mr. Brisley was a member of the Chi Psi fraternity and of the American Institute of Electrical Engineers. His death occurred at his home at Wayne, Delaware, January 8, 1908.

WILLIAM HIGGINS HUME

William Higgins Hume was born in Polmont, Sterlingshire, Scotland, on July 12, 1877. He came to America with his parents in 1883, residing a few years in South Carolina. Later they removed to Troy, Alabama, where at the Troy Normal College Mr. Hume received a part of his education.

In 1894 he went to Scotland and studied two years in the Herriott College, Edinburgh. Upon his return to America in 1896, he entered the machine shop and office of the Standard Chemical and Oil Company of Troy, Alabama. In 1898 he was draftsman with the Bethlehem Steel Company and had later engagements with Jones & Laughlin in Pittsburg and the Edgemore Iron Company at Wilmington.

In 1902 he became Superintendent of Construction of the Georgia Iron and Coal Company at Rising Fawn, Georgia, during the rebuilding of their blast furnace plant, and later was chief engineer of the same company.

In 1904 he was general sales manager of the Herron-Brady Pump and Foundry Company, Chattanooga, Tenn., leaving there in 1907 to accept the appointment of superintendent of foundry of the Bucyrus Company of South Milwaukee, Wisconsin. He held this position until his death, which occurred January 3, 1908.

CHARLES H. L. SMITH

Charles H. L. Smith was born in New York, November 14, 1843. He attended public and private schools, and for three years was a student at Rensselaer Polytechnic Institute. He studied architecture and was engaged as architectural draftsman in the office of Thomas C. Smith, at the same time studying mechanics and machinery. In 1876 he became associated with his father in the manufacture of porcelain, the firm name being the Union Porcelain Works, located at Greenpoint, New York City, the only pottery in the United States where true hard porcelain is manufactured. It also has the distinction of being the only factory to produce hard china without the government aid received by factories abroad. During his connection with this firm Mr. Smith introduced the hard kaolinic body and invented a method which made it possible to make an oval dish by machinery. He accomplished this by applying the eccentric principle to the potter's wheel. He designed a great deal of machinery for use in his plant.

Mr Smith was a member of the Masonic Order, life member of the New York Historical Society, the American Institute of Arts and Sciences, the New England Society; a member of the Manufacturers Association of New York, the New York Board of Trade and Transportation, the Garden City Golf Club and the Crescent Athletic Club. Mr. Smith died March 6, 1908.

W. H. WIGGIN

W. H. Wiggin was born at Dracut, Massachusetts, May 7, 1861. He attended the public school at Chelmsford and the Academy at New Hampton, graduating in 1879.

His apprenticeship was served in a repair shop at Greenville, New Hampshire, and later he was employed by the Fitchburg Machine Company, the Putnam Machine Company, and various other shops

at Fitchburg, Mass. In 1883 he was given charge of the Lamson Cash Railway Company, Lowell, Massachusetts, and at this time took out a patent for improvements on cash railway systems for stores. In 1886 he was employed by Charles H. Morgan of Worcester, Masschusetts, on special work in developing new machinery. A year later he was engaged as mechanical man in a factory manufacturing medical capsules, afterwards returning to C. H. Morgan, at Worcester, where he did erecting work on rolling mills, and designed new machinery.

Mr. Wiggin was subsequently engaged with G. L. Brownell, and the H. C. Pease Machine Company, the old Washburn & Moen Manufacturing Company, Marcus Mason and Company and the Richardson Manufacturing Company and at this time designed a new mower for the rocky hillside farms of New England. In 1903 he became mechanical assistant superintendent at the Deering Division of the International Harvester Company, and was advanced to the position of master mechanic in the construction and equipment division. He remodeled and reëquipped the old Milwaukee Harvester Company plant for the manufacture of gasolene engines and cream separators. In 1905 he was transferred to Hamilton, Ontario, as superintendent of the Canadian plant, and held this position until his death, October 2, 1907.

Among Mr. Wiggin's most important inventions was a rail bond which is used on electric roads and an automatic machine for making bicycle spokes and similar products. He was a member of the Masonic Order, and of the Canadian Manufacturers Association.

ANNOUNCEMENT

Under the direction of the Council the Committee on Society History has arranged to present the results of its investigations to the members of the Society.

The Preliminary Report will appear in the proceedings of the Society from month to month, and thus enable the matter to be open to comment during its completion. It is especially desired that any member who may be in the possession of facts or information bearing upon the various points as they are thus made public will communicate with the committee, in order that the final and completed report may have the advantage of the collaboration of the membership at large.

HISTORY OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PRELIMINARY REPORT OF THE COMMITTEE ON SOCIETY HISTORY

CHAPTER 3

SOCIAL DEVELOPMENTS

85 One of the interesting features accompanying the early progress of the Society appears in the fact that from the start it became identified with the social relations of the members to each other as well as with their professional work.

86 At the time the Society was founded, the principal form of social entertainment for professional and other organizations was that of a dinner, and it is interesting to note that a dinner was held at the Astor House on the evening of February 16, 1880, the same day on which the preliminary meeting had been held for the formation of the Society. Fortunately one of the original menu cards of this dinner has been preserved and is in the possession of the Society, having been placed in the hands of the Committee by Mr. Fred J. Miller, of the *American Machinist*. That the name of the Society had not yet

been definitely determined appears by the heading of this card, which reads:

COMPLIMENTS

OF THE

AMERICAN MACHINIST

TO THE

MECHANICAL ENGINEERS' SOCIETY

87 Referring to this dinner, Professor Sweet writes:

The preliminary meeting was held at the office of the *American Machinist*, whose editors took great interest in the formation of the Society, and invited those at the meeting to join them in a banquet at the Astor House that evening.

There must have been 20 or 25 present. Horace B. Miller took the seat at the head of the table, and Charles E. Emery and E. D. Leavitt helped me brace him up. Almost every one was called upon for remarks, and, as may well be believed, Mr. Holley was the one well worth listening to. I shall never forget the right and left hand honors conferred upon me that night by the men at the table, and by the proprietors of the Astor House, where I felt bound to put up.

One of the earliest humorous books which I recollect was that by Sam Slick, a Connecticut Yankee, who traveled the country over, selling Yankee clocks, wooden nutmegs, and basswood pumpkin seeds. In recounting his travels he tells how he patronized the Astor House when in New York, and said that they gave him a room with a "stun wreath around the winder," and on this occasion the hotel people gave me the same room. In his day it faced Barnum's Museum at the corner of Park Row, where today stands the St. Paul building. The window of that room still has the same "stun wreath" around it.

As the original meeting at the old offices of the *American Machinist* was smaller, but more important, than all that have followed, so that banquet, the smallest we have ever had, was as happy as any we have since held.

88 Thus the Society, from its early beginnings, combined social with professional gatherings, and established a precedent for its meetings which gave it a status altogether different from meetings of mechanics or organizations of manufacturers.

89 The second organization meeting, held on April 7, 1880, at the Stevens Institute at Hoboken, held no evening gathering, but as its deliberations occupied the greater part of the day, there was opportunity for an entertainment of a different kind, the members being the guests of President Henry Morton at luncheon, at his residence. This provided a delightful rest and social diversion in the middle of a day which, as has already been recorded, was devoted to active discussion of the fundamental business organization of the Society.

90 At the first few meetings after organization the social feature of the gathering continued to be the dinner, this being a subscription affair for those who felt inclined to attend. To this, however, there was added, at meetings in industrial centers, visits to shops, manufacturing establishments, and the like. Thus, at the Hartford meeting, in May 1881, visits were made to the works of the Pratt & Whitney Company, Billings & Spencer Company, and to the Colt Fire Arms Manufacturing Company, while Trinity College, an educational institution of high standing, although not devoted to engineering instruction, made the members of the Society most welcome.

91 At the Altoona meeting, held in August 1881, the formal social affair, as heretofore, was a subscription dinner, but the great objects of professional interest were the shops of the Pennsylvania Railroad Company, at Altoona, and the works of the Cambria Company, at Johnstown, at both of which important establishments the members were made most welcome.

92 The meetings in New York City have always had to provide for the fact that many of the members from other parts of the country desire time to attend to business and social affairs of their own, and the conduct of the New York meetings has included such an arrangement of sessions as would meet this requirement. Still there has always been some marked social feature of the annual meeting, bearing out the fact that the Society was an organization of a high personal as well as professional character, a position which has always continued to mark its career.

93 The feature of the New York meeting held in November 1881 was again a subscription dinner, this function being held at Delmonico's, and on this occasion the presence of ladies was included, thus leading the way to the development of the reception which subsequently became the recognized form of entertainment.

94 The first occasion upon which the formal reception became the social function of the meetings of the Society, however, was at the Philadelphia meeting, in April 1882.

95 At this meeting especial efforts were made by the local committee to provide for the Society a welcome both of a professional and social character which should, so to speak, set the pace for the future growth of the organization. This was made possible, both by the coöperation of the Franklin Institute, and by the active work of a number of eminent citizens, who, although not members of the Society, responded most liberally with services and funds.

96 The principal social feature of this meeting was an elaborate

evening reception held at the Philadelphia Academy of Fine Arts, and reported at the time as being a "splendid affair," at which about one thousand ladies and gentlemen were present. The reception committee consisted of such prominent citizens of Philadelphia as Mr. George B. Roberts, Mr. Anthony J. Drexel, Mr. George W. Childs, Dr. William Pepper, Hon. George H. Boker, and Mr. Fairman Rogers, there being thus represented the leading railway, banking, publishing, literary, and educational interests, besides the engineering profession. The active spirits in the conduct of the social side of the Philadelphia meeting, besides the gentlemen above mentioned were Messrs. Sellers, Baldwin, Bement, and other members of the Society, and there is no doubt that from this time the conventions took on more of a social character, without in any way interfering with the continual progress of the professional and technical side of the work.

97 The influence of the Philadelphia meeting of 1882 upon the future progress of the Society appeared in the meeting held at Cleveland, Ohio, in the following summer. As has already been noted, this was the first meeting held west of the Atlantic coast, and especial efforts were made to give to it such attractions as would show that the Society was really a national organization, and that it could receive as hearty a welcome at Cleveland as it had received at any previous place of meeting.

98 The idea of holding a Cleveland meeting originated with Mr. J. F. Holloway, and as soon as the Society decided to accept the invitation a local committee of arrangements was appointed, consisting of J. F. Holloway, Charles F. Brush, W. M. Barr, Ambrose Swasey, N. S. Parsons, W. H. Thompson, S. T. Wellman, E. H. Martin, John Walker, W. R. Warner, J. D. Cox, and F. H. Richards. It will be seen that, as in the case of Philadelphia, the committee consisted, not only of active members of the Society, but also of prominent residents of the city, all determined upon making the convention a great success, not only from the professional but also from the social point of view. Mr. Holloway, as chairman of the local committee, visited many of the influential and fashionable people of Cleveland, and showed them that this was no ordinary gathering of mechanics, but that it was the opportunity of the city to welcome some of the most eminent professional men of the country, the men who had conceived and planned its railways, canals, and manufactories; the men who were largely directing the industrial energies of the nation in its advancing civilization.

99 His mission was highly successful, and he succeeded in obtain-

ing coöperation and ample subscriptions for the entertainment of the coming engineers.

100 The convention was opened by an address of welcome from Hon. John H. Farley, Mayor of Cleveland, and at the conclusion of the first morning session came the timely telegram announcing the fact that the degree of Doctor of Engineering had been conferred upon President Leavitt by the Stevens Institute of Technology.

101 The professional sessions included visits to the Otis Steel Company, where, among other things, attention was directed to the Porter-Allen engine of 40 in. bore and 48 in. stroke, running at 100 revolutions, and developing 1500 h. p. This engine was one of the early high-speed engines, and the late Captain Jones, when visiting Mr. Wellman, then manager of the Otis Steel Works, said that anyone running such an engine at such a speed ought to be arrested for cruelty to machinery. The engine continued to run at the rate of 100 r. p. m. for more than twenty-five years, operating a three-high train mill of the Lauth type, with rolls 112 in. long and 30 in. in diameter, rolling plates 105 in. wide and 12 ft. long. This was the largest three-high train plate-mill which had been built up to that time, and the whole plant attracted much attention as being the most advanced methods in rolling steel plate then constructed.

102 At the works of Warner & Swasey the members of the Society were especially interested in the running gear for the 45 ft. dome of the observatory of Virginia, this being for a telescope 26 in. in diameter. Such domes had previously required so much power to revolve them that it was necessary to use either a motor or gearing, but with the mechanism of this dome it was found possible to revolve it with a direct pressure of but 1½ lb., so that no gearing whatever was required.

103 On the evening of June 13 there was held a reception, tendered to the Society by the citizens of Cleveland; the use of the Opera House, where the reception was held, having been contributed by Mr. M. A. Hanna, the owner, at that time largely interested in iron mining and in many of the industrial enterprises of the vicinity, and afterwards an important figure in national politics. Mr. Hanna was one of the prominent citizens who showed an intense interest in the convention, and took a very active part, with others, in the entertainment of the Society.

104 Such a development of the social side of the convention work as was thus seen at the Philadelphia and Cleveland meetings naturally had its influence upon subsequent gatherings of the Society, but

the progress was by no means uniform. At the New York meetings, as has already been mentioned, the general attractions of the metropolis demanded a portion of the time of the visiting members, and for a time the social feature was still limited to the subscription dinner, and this was true also at the meeting in Pittsburg, in May 1884, and at Atlantic City, in May 1885.

105 In November 1885 a departure was made in the practice of the Society and the Annual meeting was held in Boston, instead of New York City. Here again the social features of the meeting were developed in a most successful manner, a brilliant reception being tendered to the Society by the Boston Art Club in its own club house.

106 As a comment upon the conduct of the social features of the Boston meeting, the following extract from one of the resolutions passed at the close of the Convention will be found of interest:

> One of the finest pieces of engineering which has come under our favored notice in the charming city of Boston, has been that of Mr. C. J. H. Woodbury and the local committee. They have solved the problem how some 400 feet of mechanical engineers can be successfully run at varying speeds in a city with bearings very much out of line, with the least possible amount of friction, say, one-half of 1 per cent.

107 At the Chicago meeting of 1886 a banquet was given to the visiting members by the local committee, and at the Washington meeting in the spring of 1887 the Honorable Josiah Dent and Mr. Edward L. Dent gave an especially noteworthy reception to the Society, this being remarkable as a social function tendered to the members in the private residence of one of its members.

108 The social element had now taken a firm hold upon the conduct of the meetings. This is well shown by the manner in which the second Philadelphia meeting was carried out, in November 1887, this being another occasion on which the annual meeting was held elsewhere than in New York City.

109 Animated by a desire to surpass both the former Philadelphia meeting and the Cleveland meeting, the public-spirited local committee raised the sum of $10 000 for the expenses of the occasion, contributions being made by many of the leading manufacturing and engineering firms in the city. The management of the affair was in the hands of the prominent business and social leaders of Philadelphia, and the use of the Academy of Fine Arts was secured for the reception. Opportunity was also offered to the members to visit the large collection of art treasures in this historical art museum, a

collection which owes its existence largely to the interest given to its foundation by Benjamin West and by his pupil Robert Fulton.

110 The custom of holding such receptions, attended by the wealth and fashion of the city where the convention was held, thus became well established, and firmly fixed the social standing of the meetings of the Society, and the precedents created in this manner were still more effectively impressed by the international relations, both professional and social, which followed upon the European trip of 1889.

(To be continued)

SOME PITOT TUBE STUDIES

The Distribution of Velocities and Pressures in Straight and Curved Portions of a Six-Inch Water Pipe

By PROF. W. B. GREGORY, NEW ORLEANS, LA.
Member of the Society

and PROF. E. W. SCHODER,[1] ITHACA, NEW YORK
Non-Member

It is the nature of science that lump effects should be the first to be observed and studied. Later there come the more and more detailed investigations that make for an exact science. To this manner of evolution hydraulics is no exception.

2 In the design of water wheels, pumps, air blowers and gas engines, the improvements have come as the result of numberless tests involving accurate measurements of total quantities of discharge or supply and therefore of mean velocity of the liquid or gas. Such measurements may be *direct* as in the case of tank measurements, of total discharge in a given time, or indirect as in the case of measurement by a Venturi meter or Pitot tube. However, if we desire a knowledge of the *distribution* of velocities and pressures in a pipe or passage as well as of the total flow, then the Pitot tube is the only instrument available.

3 The writers have made a study of the distribution of velocities in a six-inch pipe by means of the Pitot tube, undertaken in connection with experiments on the loss of head due to elbows of varying radius. However, this paper does not discuss the loss of head in elbows. It was desired to know how far down the pipe the distortion

[1] In charge of Hydraulic Laboratory, Cornell University.

To be presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

The professional papers contained in Proceedings are published prior to the meetings at which they are to be presented, in order to afford members an opportunity to prepare any discussion which they may wish to present. They are issued to the members in confidence, and with the understanding that they are not to be published even in abstract, until after they have been presented at a meeting. All papers are subject to revision.

The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions.—C55.

in velocities due to the presence of the curve extended, or more exactly, to know whether there was elliptical distribution of velocities at distances of approximately 168 diameters and 76 diameters beyond the curve.

4 The distribution of velocities in the curve was also investigated. Traverses were made with a Pitot tube both in the plane of the curve and across the pipe at right angles with this plane. And a method was developed whereby the pressure at any point in a pipe, whether straight or curved, may be obtained by means of the Pitot tube.

FIG. 1 VIEW OF CORNELL UNIVERSITY HYDRO-ELECTRIC POWER PLANT, SHOWING SIX-INCH PIPE LINE USED IN THE PITOT TUBE EXPERIMENTS

5 The arrangement of the pipe line is clearly shown in Fig. 1 and Fig. 2. The pipe line was set up in Fall Creek Gorge, near the Cornell Hydro-Electric Power Plant, so that the water supply could be taken from the blow-off end of the five-foot steel supply pipe where a head of approximately 135 ft. is available.

6 The curve in which the investigations with the Pitot tube were made was a 90 deg. circular elbow with radius of center line 2.5 ft., or five diameters. It was bent from six-inch wrought iron pipe. The ends were threaded for ordinary flanges.

7 The Pitot tube used is shown in place in Fig. 3 and Fig. 4. The details of the tube are shown in Fig. 6.

8 Traverses were made in both horizontal and vertical planes at points on the pipe 1.10 ft. upstream from the curve, at approximately 22½, 45 and 67½ deg. in the curve and also at points in the straight pipe beyond the curve distant 84.10 ft. and 38.0 ft., respectively. See Fig. 2, 3 and 4.

9 In every case where a traverse was made in straight portions of the pipe line the static pressure at the wall of the pipe was obtained from two diametrically opposite openings in the same cross section as

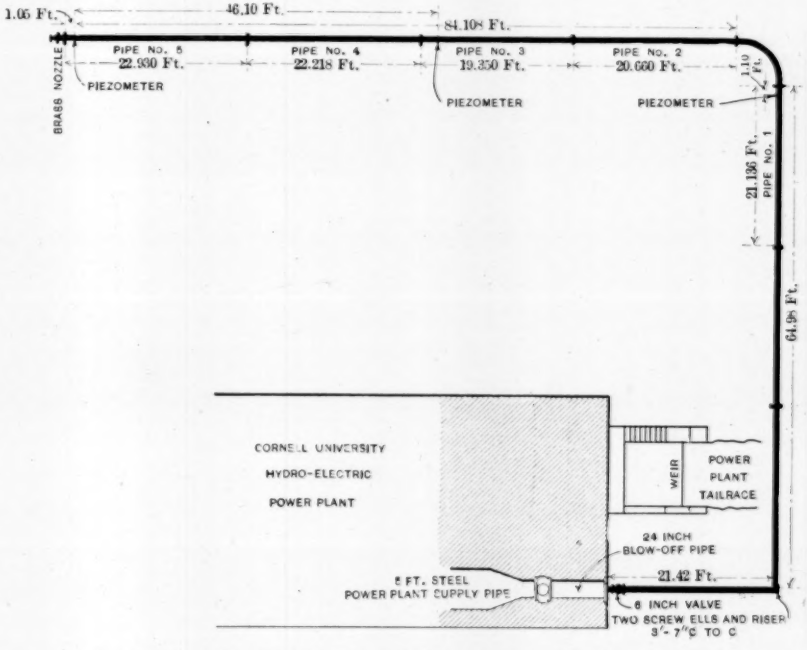

FIG. 2 PLAN OF SIX-INCH WROUGHT IRON PIPE LINE

the Pitot tube and at right angles to the diameter on which the traverse was made, These holes were tapped out to receive one-eighth inch brass cocks. After drilling and tapping each hole the burr was carefully removed from the inside of the pipe with a file, and care was taken that the cocks should not extend beyond the inner wall of the pipe when screwed home.

10 For all traverses in the curve the static pressures were obtained from the two horizontal openings 1. 10 ft. upstream from the curve. In making traverses, readings were obtained at one-half inch intervals

FIG. 3 PITOT TUBE IN POSITION FOR HORIZONTAL TRAVERSE IN STRAIGHT PIPE AND (AT THE LEFT) THE DIFFERENTIAL GAGE

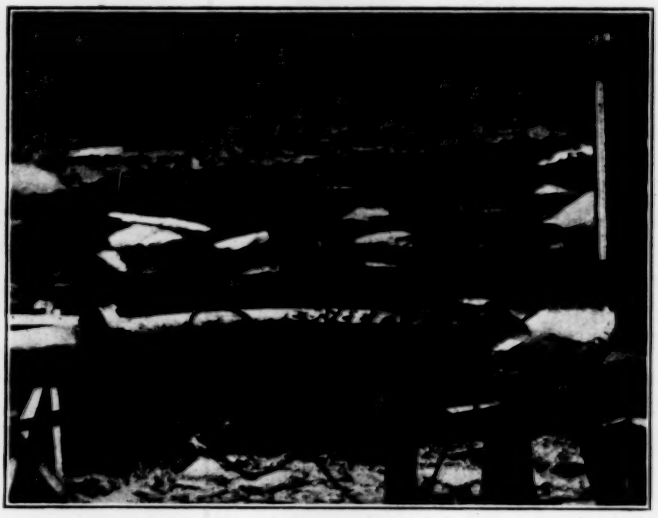

FIG. 4 PITOT TUBE IN POSITION FOR VERTICAL TRAVERSE IN CURVE. HOSE CONNECTIONS AND DIFFERENTIAL GAGE

in some cases, while in others the points were so distributed that the arithmetical mean could be used directly in getting mean velocity. This matter will be discussed presently.

11 In taking readings with the Pitot tube the instrument was carefully adjusted at the desired point on the diameter to be traversed, with the point opening facing the current. The tube was connected to one side of the differential water gage shown in Fig. 3 and Fig. 4, and the wall openings in the pipe were connected to the other side. All connections were made by means of small three-ply rubber hose. Great care was used to blow off all air from the hose connections. After the readings with the tube direct (i. e., pointing upstream), were taken, the tube was reversed to point downstream and another set of readings obtained.

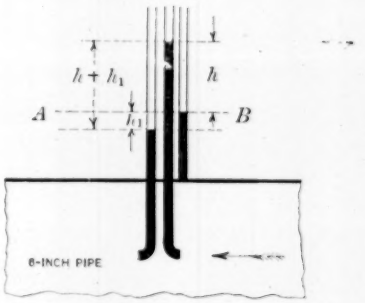

FIG. 5 DIAGRAMMATIC REPRESENTATION OF PITOT TUBE DIRECT AND REVERSE

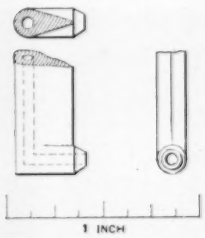

FIG. 6 DETAIL OF PITOT TUBE

12 As a check on the Pitot tube measurements the quantity of water was measured by means of a brass nozzle at the end of the pipe line. The nozzle had been calibrated previously by tank measurement in the Cornell University Hydraulic Laboratory. To read the head on the nozzle a U tube mercury gage, open to the atmosphere on one side, was used.

13 By using the observed water gage difference when the point of the tube was facing the current as h, in the formula $v = c\sqrt{2gh}$, it was found that the value of c is unity, within a reasonable limit of error, as will be seen from the following table.

1	2	3	4	5	6
Date	Place of traverse	Mean velocity from point direct-wall reading. Pitot tube	Mean velocity from nozzle	Col. 3 / Col. 4	Ratio mean to center velocity. Pitot tube
Oct. 28, 1907	Vertical, 1.10 ft. upstream from curve.	16.66	16.73	0.996	0.8355
Oct. 29, 1907	Vertical, 1.10 ft. upstream from curve.	7.95	7.71	1.031	0.842
Oct. 31, 1907	Horizontal, 84.10 ft. downstream from curve	7.93	7.78	1.019	0.843
Nov. 4, 1907	Horizontal 84.10 ft. downstream from curve	8.39	8.24	1.018	0.850
Nov. 5, 1907	Horizontal, 38.0 ft. downstream from curve	7.83	7.92	0.989	0.832
	Mean			1.011	0.8406

14 The results indicate an average error of about 1 per cent while the greatest error in any case is about 3 per cent. As variations as wide as the above may be expected in work of this kind, even when conducted with the greatest care, the conclusion that the coefficient

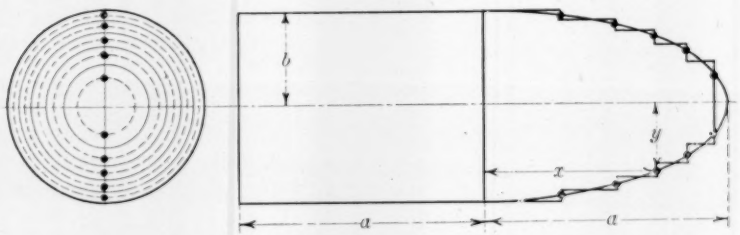

FIG. 7 DIAGRAM SHOWING POSITIONS OF PITOT TUBE FOR TEN POINT METHOD OF TRAVERSE

of the tube is unity is justified. Probably a greater number of traverses would have given an average still nearer to unity.

The plottings of the traverses in the straight pipe are shown in Fig. 8.

15 Normal flow is shown by the shape of the curves and the ratios of mean to center velocities. See preceding table. The exact distance downstream from the curve where the distortion of velocities disappears is not known.[1] But it is shown by the traverse of November

[1] Saph and Schoder, Trans. Am. Soc. C. E., vol. 47, 1902, pp. 300–302.

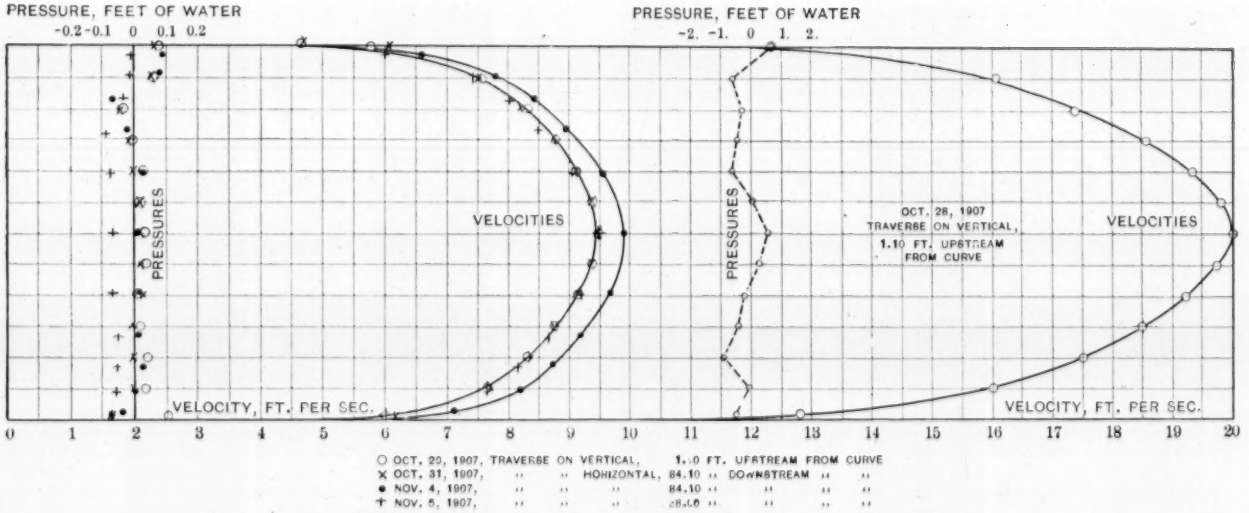
PRESSURE, FEET OF WATER
-0.2 -0.1 0 0.1 0.2
PRESSURES
VELOCITIES
VELOCITY, FT. PER SEC.
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
PRESSURE, FEET OF WATER
-2. -1. 0 1. 2.
PRESSURES
OCT. 28, 1907
TRAVERSE ON VERTICAL,
1.10 FT. UPSTREAM
FROM CURVE
VELOCITIES
VELOCITY, FT. PER SEC.
○ OCT. 29, 1907, TRAVERSE ON VERTICAL, 1.10 FT. UPSTREAM FROM CURVE
× OCT. 31, 1907, " " HORIZONTAL, 84.10 " DOWNSTREAM " "
● NOV. 4, 1907, " " " 84.10 " " " "
† NOV. 5, 1907, " " " 28.06 " " " "

FIG. 8 DISTRIBUTION OF VELOCITIES AND PRESSURES IN A STRAIGHT PIPE

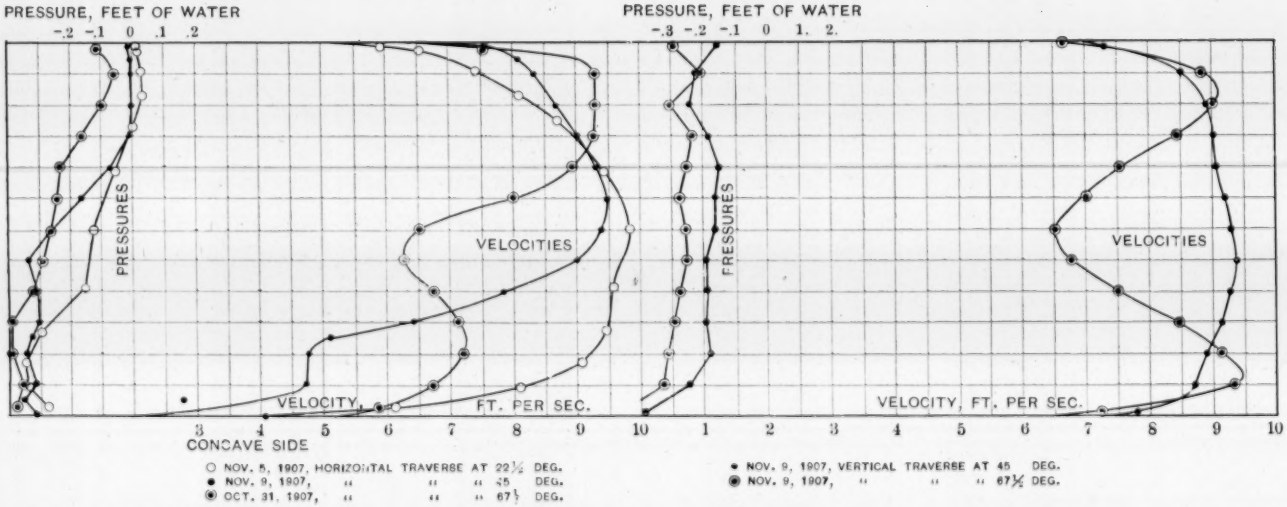
PRESSURE, FEET OF WATER
-.2 -.1 0 .1 .2
PRESSURES
VELOCITIES
VELOCITY, FT. PER SEC.
3 4 5 6 7 8 9 10
CONCAVE SIDE
PRESSURE, FEET OF WATER
-.3 -.2 -.1 0 1. 2.
PRESSURES
VELOCITIES
VELOCITY, FT. PER SEC.
1 2 3 4 5 6 7 8 9 10
○ NOV. 5, 1907, HORIZONTAL TRAVERSE AT 22½ DEG.
● NOV. 9, 1907, " " " 45 DEG.
◉ OCT. 31, 1907, " " " 67½ DEG.
• NOV. 9, 1907, VERTICAL TRAVERSE AT 45 DEG.
◉ NOV. 9, 1907, " " " 67½ DEG.

FIG. 9 DISTRIBUTION OF VELOCITIES AND PRESSURES IN A CURVED PIPE

5, 1907, that the flow is normal at a distance of 38 ft., or 76 diameters below the curve.

16 The gage readings obtained by reversing the tube have been investigated in connection with their relations both to the velocity and the pressure of the water at the various parts of the cross section of the pipe. The theory may be stated as follows:

17 At the right of Fig. 5 the Pitot tube is shown facing the current, while on the left the reversed position of the tube is shown. As already stated, a differential gage was used to read the quantities h and h_1, but for the sake of clearness, let open piezometers be imagined, so that the level of water as shown by the wall piezometer will be at the constant level A–B. As both h and h_1 are differences, it does not matter whether open piezometers or differential gages are used.

18 Let v be the true velocity for any point as obtained by the Pitot tube, point direct-wall reading.

Then $v = \sqrt{2gh}$

$v_1 = \sqrt{2g(h + h_1)}$, where h_1 = suction effect due to reversing the tube.

$$v_1^2 = 2gh + 2gh_1 = v^2 + 2gh_1$$

Therefore,

$$h_1 = \frac{v_1^2 - v^2}{2g}$$

But, from experiment,

$$v_1 = 1.133\,v$$

Therefore,

$$h_1 = \frac{(1.133^2 - 1)\,v^2}{2g} = \frac{(1.284 - 1)\,v^2}{2g} = 0.0044\,v^2$$

19 The value 1.133 is the mean obtained from the five straight pipe traverses. The individual results for each point of the traverses are given in Table 1. A summary follows:

Date	Average value of $c = \frac{v_1}{v}$
October 28, 1907	1.131
October 29, 1907	1.115
October 31, 1907	1.131
November 4, 1907	1.128
November 5, 1907	1.161
Mean	1.133

TABLE 1

VERTICAL TRAVERSE 1.10 FT. UPSTREAM FROM CURVE. A DIFFERENTIAL MERCURY GAGE WAS USED. THE DIFFERENCES IN THE TABLE ARE THE EQUIVALENT WATER DIFFERENCES

ENTRY ON TOP OF PIPE — OCTOBER 28, 1907

1	2	3	4	5	6	7	8	9
Distances in inches from bottom of pipe	Wall-point direct difference = h	Wall-point reversed, difference = h_1	$h + h_1$	$v = \sqrt{2gh}$	$v_1 = \sqrt{2g(h+h_1)}$	Ratio $\frac{v_1}{v}$	$0.0044\,v^2$	Col. 8 Col. 3
$\frac{1}{16}$	2.55	1.22	3.77	12.80	15.40	1.203	0.721	−0.50
$\frac{1}{2}$	3.97	1.27	4.24	16.00	16.52	1.032	1.127	−0.14
1	4.75	2.26	7.01	17.50	21.26	1.215	1.350	−0.91
$1\frac{1}{2}$	5.32	1.97	7.29	18.50	21.65	1.170	1.510	−0.46
2	5.75	1.90	7.65	19.22	22.20	1.155	1.630	−0.27
$2\frac{1}{2}$	6.04	1.49	7.53	19.72	22.02	1.117	1.710	+0.22
3	6.20	1.20	7.40	20.00	21.85	1.092	1.760	+0.56
$3\frac{1}{2}$	6.09	1.68	7.77	19.80	22.38	1.128	1.725	+0.04
4	5.80	2.33	8.13	19.32	22.90	1.185	1.640	−0.69
$4\frac{1}{2}$	5.34	2.00	7.34	18.55	21.75	1.172	1.520	−0.48
5	4.65	1.65	6.30	17.32	20.15	1.164	1.320	−0.33
$5\frac{1}{2}$	4.01	1.75	5.76	16.05	19.27	1.200	1.130	−0.62
6	2.36	−0.06	2.30	12.31	12.17	0.987	0.665	+0.60

VERTICAL TRAVERSE 1.10 FT. UPSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON TOP OF PIPE — OCTOBER 29, 1907

$\frac{1}{16}$	0.554	0.051	0.605	5.97	6.25	1.048	0.157	+0.106
$\frac{1}{2}$	0.802	0.223	1.125	7.62	8.50	1.115	0.255	+0.032
1	1.064	0.261	1.325	8.27	9.22	1.115	0.301	+0.040
$1\frac{1}{2}$	1.188	0.320	1.508	8.73	9.85	1.115	0.336	+0.016
2	1.295	0.360	1.655	9.12	10.32	1.133	0.366	+0.006
$2\frac{1}{2}$	1.365	0.350	1.715	9.36	10.50	1.120	0.386	+0.036
3	1.395	0.363	1.758	9.46	10.63	1.124	0.394	+0.031
$3\frac{1}{2}$	1.369	0.373	1.742	9.37	10.59	1.130	0.387	+0.014
4	1.290	0.338	1.628	9.10	10.23	1.126	0.364	+0.026
$4\frac{1}{2}$	1.198	0.347	1.545	8.77	9.97	1.137	0.338	−0.009
5	1.073	0.342	1.415	8.30	9.53	1.150	0.303	−0.039
$5\frac{1}{2}$	0.887	0.195	1.082	7.56	8.34	1.116	0.252	+0.057
6	0.515	0.072	0.587	5.76	6.15	1.069	0.146	+0.074

TABLE 1—Continued

HORIZONTAL TRAVERSE 1.05 FT. UPSTREAM FROM NOZZLE AND 84.11 FT. DOWNSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON LEFT HAND SIDE, LOOKING DOWNSTREAM — OCTOBER 31, 1907

1	2	3	4	5	6	7	8	9
Distance in inches from bottom of pipe	Wall-point direct difference $= h$	Wall-point reversed, difference $= h_1$	$h + h_1$	$v = \sqrt{2gh}$	$v_1 = \sqrt{2g(h + h_1)}$	Ratio $\frac{v_1}{v}$	$0.0044\ v^2$	Col. 8 Col. 3
$\frac{3}{4}$	0.588	0.238	0.776	6.15	7.07	1.150	0.166	−0.072
$\frac{1}{2}$	0.915	0.268	1.183	7.67	8.72	1.146	0.259	−0.009
1	1.080	0.308	1.388	8.32	9.45	1.135	0.305	−0.003
$1\frac{1}{2}$	1.195	0.342	1.537	8.76	9.94	1.133	0.338	−0.004
2	1.289	0.343	1.632	9.10	10.24	1.127	0.365	+0.022
$2\frac{1}{2}$	1.365	0.375	1.740	9.36	10.59	1.130	0.386	+0.011
3	1.392	0.380	1.772	9.45	10.68	1.130	0.393	+0.013
$3\frac{1}{2}$	1.362	0.372	1.734	9.36	10.54	1.127	0.386	+0.014
4	1.282	0.370	1.652	9.07	10.30	1.135	0.362	−0.008
$4\frac{1}{2}$	1.187	0.360	1.547	8.73	9.99	1.144	0.335	−0.025
5	1.043	0.343	1.386	8.18	9.43	1.152	0.295	−0.048
$5\frac{1}{2}$	0.880	0.198	1.078	7.52	8.31	1.105	0.248	+0.050
6	0.575	0.102	0.677	6.08	6.60	1.085	0.163	+0.061

HORIZONTAL TRAVERSE 1.05 FT. UPSTREAM FROM NOZZLE AND 84.11 FT. DOWNSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON LEFT HAND SIDE, LOOKING DOWNSTREAM — NOVEMBER 4, 1907.

TEN POINT METHOD. POINT NO. 1 IS OPPOSITE ENTRY

1	0.782	0.260	1.042	7.10	8.19	1.154	0.222	−0.038
2	1.041	0.295	1.336	8.19	9.27	1.133	0.295	0.000
3	1.180	0.305	1.485	8.72	9.77	1.118	0.334	+0.029
4	1.310	0.361	1.671	9.18	10.38	1.132	0.372	+0.011
5	1.452	0.408	1.860	9.67	10.94	1.133	0.411	+0.003
center	1.525	0.422	1.947	9.90	11.20	1.132	0.431	+0.009
6	1.417	0.375	1.792	9.54	10.74	1.129	0.400	+0.025
7	1.241	0.375	1.616	8.94	10.20	1.142	0.352	−0.023
8	1.095	0.380	1.475	8.40	9.75	1.160	0.311	−0.069
9	0.930	0.190	1.120	7.74	8.50	1.100	0.263	+0.073
10	0.671	0.105	0.776	6.58	7.07	1.074	0.191	+0.086

TABLE 1—Continued

HORIZONTAL TRAVERSE 38.0 FT. DOWNSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON LEFT HAND SIDE, LOOKING DOWNSTREAM NOVEMBER 5, 1907

TEN POINT METHOD. POINT NO. 1 IS OPPOSITE ENTRY

1	2	3	4	5	6	7	8	9
Distance in inches from bottom of pipe	Wall-point direct difference $= h$	Wall-point reversed, difference $= h_1$	$h + h_1$	$v = \sqrt{2gh}$	$v_1 = \sqrt{2g(h+h_1)}$	Ratio $\frac{v_1}{v}$	$0.0044\,v^2$	Col. 8 Col. 3
1	0.558	0.230	0.788	6.00	7.12	1.186	0.158	−0.072
2	0.902	0.314	1.216	7.62	8.85	1.162	0.255	−0.059
3	1.033	0.344	1.377	8.15	9.42	1.155	0.292	−0.052
4	1.170	0.382	1.552	8.68	10.00	1.150	0.332	−0.050
5	1.307	0.439	1.746	9.17	10.62	1.156	0.370	−0.069
center	1.388	0.459	1.847	9.45	10.90	1.154	0.393	−0.066
6	1.262	0.433	1.695	9.02	10.45	1.158	0.358	−0.075
7	1.115	0.404	1.519	8.47	9.88	1.167	0.316	−0.088
8	1.000	0.422	1.422	8.02	9.57	1.194	0.283	−0.039
9	0.851	0.257	1.108	7.41	8.45	1.139	0.242	−0.015
10	0.557	0.190	0.747	5.99	6.93	1.155	0.180	−0.010

20 It will be seen from the calculated values for $\frac{v_1}{v}$ in Table 1, that for any particular traverse the ratio is practically constant across the pipe, except near the wall where eddies and disturbances tend to vitiate observations.

21 If now we calculate backwards and subtract from h_1 the quantity $0.0044v^2$, we will get zero on the average. The variation from zero for the five straight pipe traverses is shown in Table 1 and in Fig. 8 at the left.

22 This average value of zero across the pipe diameter means that the pressure throughout the cross section is identical with that at the wall (the effect of hydrostatic differences of level, of course, excepted).

23 The quality of pressure throughout the cross section of a straight pipe in which water is flowing may be inferred from the fact that a Pitot tube having a point opening only, and taking its reference pressure at the wall of the pipe, gives the true mean velocity according to the formula,

$$v = \sqrt{2gh}$$

TABLE 2

HORIZONTAL TRAVERSE, IN CURVE AT 45 DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON CONCAVE SIDE — NOVEMBER 9, 1907

1	2	3	4	5	6	7	8
Distance in inches from side opposite entry	Wall-point direct difference $= h$	Wall-point reverse difference $= h_1$	$h + h_1$	$v = \frac{v_1}{1.133} = 0.883\, v_1$ $= 0.883\sqrt{2g(h+h_1)}$	$v_1 = \sqrt{2g(h+h_1)}$	$0.0044\, v^2$	Col. 7 / Col. 3
$\frac{1}{16}$	0.870	0.271	1.141	7.58	The ratio $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.253	−0.018
$\frac{1}{8}$	1.005	0.293	1.298	8.08		0.288	−0.005
$\frac{1}{2}$	1.080	0.300	1.380	8.33		0.306	−0.006
1	1.170	0.335	1.505	8.68		0.333	−0.002
$1\frac{1}{2}$	1.250	0.372	1.622	9.03		0.360	−0.012
2	1.277	0.453	1.730	9.31		0.382	−0.071
$2\frac{1}{2}$	1.223	0.563	1.786	9.48		0.396	−0.167
3	1.103	0.653	1.756	9.39		0.389	−0.264
$3\frac{1}{2}$	0.920	0.695	1.615	9.00		0.357	−0.338
4	0.644	0.583	1.227	7.85		0.272	−0.311
$4\frac{1}{2}$	0.340	0.485	0.825	6.44		0.183	−0.302
$4\frac{3}{4}$	0.083	0.437	0.520	5.11		0.115	−0.322
5	0.005	0.447	0.452	4.76		0.100	−0.347
$5\frac{1}{2}$	0.030	0.413	0.443	4.72		0.098	−0.315
$5\frac{3}{4}$	−0.245	0.390	0.155	2.79		0.034	−0.356
6	−0.057	0.387	0.330	4.07		0.073	−0.314

HORIZONTAL TRAVERSE, IN CURVE AT 67½ DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON CONCAVE SIDE — OCTOBER 31, 1907

$\frac{1}{8}$	0.780	0.360	1.140	7.55	The ratio $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.248	−0.112
$\frac{1}{2}$	1.280	0.441	1.721	9.30		0.381	−0.060
1	1.250	0.477	1.727	9.31		0.381	−0.096
$1\frac{1}{2}$	1.167	0.545	1.712	9.28		0.379	−0.166
2	1.012	0.583	1.593	8.94		0.352	−0.231
$2\frac{1}{2}$	0.743	0.525	1.278	8.01		0.282	−0.243
3	0.395	0.450	0.845	6.52		0.187	−0.263
$3\frac{1}{2}$	0.323	0.465	0.788	6.29		0.174	−0.291
4	0.373	0.523	0.896	6.71		0.198	−0.325
$4\frac{1}{2}$	0.400	0.613	1.013	7.13		0.223	−0.390
5	0.415	0.622	1.035	7.21		0.228	−0.394
$5\frac{1}{2}$	0.340	0.555	0.895	6.71		0.198	−0.357
$5\frac{13}{16}$	0.160	0.527	0.687	5.87		0.152	−0.375

TABLE 2—Continued

VERTICAL TRAVERSE IN CURVE, 45 DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY AT TOP OF PIPE — NOVEMBER 9, 1907

1	2	3	4	5	6	7	8
Distance in inches from side opposite entry	Wall-point direct difference $= h$	Wall-point reverse, difference $= h_1$	$h + h_1$	$v = \frac{v_1}{1.133} = 0.883\, v_1$ $= 0.883\sqrt{2g\,(h + h_1)}$	$v_1 = \sqrt{2g\,(h + h_1)}$	$0.0044\, v^2$	Col. 7 Col. 3
$\frac{1}{16}$	0.556	0.655	1.211	7.80	The ratio $\frac{v_1}{v}$ as obtained in the straight pipe line was used to compute v in the curve.	0.268	−0.387
$\frac{1}{2}$	0.936	0.580	1.516	8.71		0.334	−0.246
1	1.041	0.534	1.575	8.90		0.349	−0.185
$1\frac{1}{2}$	1.109	0.563	1.672	9.15		0.369	−0.194
2	1.147	0.568	1.715	9.28		0.379	−0.189
$2\frac{1}{2}$	1.163	0.580	1.743	9.35		0.385	−0.195
3	1.174	0.542	1.716	9.28		0.379	−0.163
$3\frac{1}{2}$	1.151	0.535	1.686	9.19		0.372	−0.163
4	1.120	0.513	1.633	9.05		0.360	−0.153
$4\frac{1}{2}$	1.077	0.540	1.617	9.01		0.357	−0.183
5	0.990	0.590	1.580	8.90		0.349	−0.241
$5\frac{1}{2}$	0.897	0.543	1.440	8.50		0.318	−0.225
$5\frac{7}{8}$	0.677	0.390	1.067	7.31		0.235	−0.155

VERTICAL TRAVERSE IN CURVE, $67\frac{1}{2}$ DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY AT TOP OF PIPE — NOVEMBER 9, 1907

$\frac{1}{16}$	0.335	0.707	1.042	7.23	$v = \frac{v_1}{1.133} = 0.883\, v_1$ The ratio of $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.230	−0.477
$\frac{1}{2}$	1.027	0.713	1.740	9.34		0.384	−0.329
1	0.973	0.680	1.653	9.11		0.365	−0.315
$1\frac{1}{2}$	0.820	0.607	1.427	8.45		0.315	−0.292
2	0.598	0.523	1.121	7.50		0.248	−0.275
$2\frac{1}{2}$	0.460	0.453	0.913	6.77		0.202	−0.251
3	0.402	0.448	0.850	6.53		0.188	−0.260
$3\frac{1}{2}$	0.487	0.493	0.980	7.01		0.216	−0.277
4	0.630	0.503	1.133	7.54		0.251	−0.252
$4\frac{1}{2}$	0.870	0.550	1.420	8.44		0.314	−0.236
5	0.960	0.660	1.620	9.01		0.357	−0.303
$5\frac{1}{2}$	1.008	0.543	1.551	8.82		0.342	−0.201
6	1.400	0.486	0.886	6.66		0.195	−0.291

TABLE 2—Continued

HORIZONTAL TRAVERSE IN CURVE, 22½ DEG. FROM UPSTREAM END OF THE CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON CONCAVE SIDE NOVEMBER 5, 1907

TEN POINT METHOD. POINT NO. 1 IS OPPOSITE ENTRY

1	2	3	4	5	6	7	8
Distance in inches from side opposite entry	Wall-point direct difference $= h$	Wall-point reverse, difference $= h_1$	$h + h_1$.	$v = \frac{v_1}{1.133} = 0.883\, v_1 = 0.883\sqrt{2g\,(h + h_1)}$	$v_1 = \sqrt{2g\,(h + h_1)}$	$0.0044\, v^2$	Col. 7 / Col. 3
1	0.685	0.165	0.850	6.54		0.188	+013.0
2	0.880	0.215	1.095	7.42		0.242	+0.027
3	1.055	0.250	1.305	8.10		0.282	+0.032
4	1.170	0.332	1.502	8.70		0.333	+0.001
5	1.315	0.447	1.762	9.41		0.350	−0.057
center	1.378	0.548	1.926	9.83		0.426	−0.122
6	1.272	0.558	1.830	9.59		0.405	−0.153
7	1.090	0.695	1.785	9.47		0.395	−0.300
8	0.935	0.710	1.645	9.09		0.364	−0.346
9	0.687	0.630	1.317	8.10		0.289	−0.341
10	0.307	0.445	0.752	6.14		0.166	−0.279

$v = \frac{v_1}{1.133} = 0.883\, V_1$

The ratio $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.

This has been shown for many different sizes of pipes and for a wide range of velocities by several experimenters. If the pressure varied across the pipe this relation would not hold. Further direct experimental proof has been given by at least two observers.[1]

24 Having calculated the value $h + h_1$, from the observations on the curve traverses, we may obtain the true velocity at any point by using the relation

$$\frac{v_1}{v} = \frac{\sqrt{2g\,(h + h_1)}}{\sqrt{2\,gh}} = 1.133$$

And having the true velocity, we calculate the corresponding suction head by using the formula

$$h_1 = 0.0044\, v^2$$

If now for each point on the traverse this calculated suction head be subtracted algebraically from the corresponding gage difference with

[1] Note by M. Bazin, Trans. Am. Soc. C. E., vol. 47, 1902, p. 248, and "The Pitot Tube," W. B. Gregory, Trans. Am. Soc. M. E., vol. 25, p. 201.

the tube reversed, we have an indication of the pressure variation across the pipe diameter. The results of the curve traverses are plotted in Fig. 9.

25 The horizontal traverses in the curve show the gradual swinging of the maximum velocity toward the convex side of the curve as the water passes around, except in the case of the traverse at 22½ deg. which shows a tendency in the opposite direction. The vertical traverses show nearly symmetrical distribution above and below the center.

26 The plottings of the pressure differences as explained above, show that the pressure is highest near the convex side of the pipe and least near the concave side. The actual quantitative results are complicated by the loss of head between the upstream piezometer and the points in the curve where the traverses were made. But this is a constant and the qualitative results are not altered. The high pressure at the convex side of the curve may be explained by the centrifugal action of the water.

27 In the case of the pressures for the vertical traverses, no corresponding change of pressure with velocity is noticeable.

28 In using the Pitot tube to obtain the mean velocity of water flowing in a straight pipe a question arises as to the number of points of observation necessary that the resulting error may be reasonably small.

29 The computations given below are based on the theory that the velocities in the various parts of a cross section of a straight pipe in which water is flowing are arranged in such a way that they form an ellipsoid of revolution, with the velocity at the center equal to twice the velocity at the walls of the pipe. On any diameter the plotting of velocities parallel to the axis of a straight pipe may be thought of as made up of a semi-ellipse joined to a rectangle of length equal to the semi-ellipse.[1]

30 Suppose the cross section of the pipe to be divided by concentric circles having areas, respectively, of 1/20, 3/20, 5/20, 7/20, 9/20, 11/20, 13/20, 15/20, 17/20 and 19/20 of the cross sectional area. If in making a traverse on a diameter, observations be taken on the circumferences of these circles, there will be 20 readings for a traverse, and the mean of the velocities found from these 20 readings will be the mean velocity of the fluid. In order to investigate this theory, let us assume an ellipse having one-half of its major and minor axes

[1] The accompanying curves confirm this assumption. For similar work, see bibliography in Engineering News, December 21, 1905.

denoted by a and b, respectively. Let x and y be the coördinates of any point on the ellipse. The results are arranged in tabular form in Table 3.

TABLE 3

No	Area	y	x
1	1/20	0.2236b	0.9747a
2	3/20	0.3873b	0.9220a
3	5/20	0.5000b	0.8660a
4	7/20	0.5916b	0.8062a
5	9/20	0.6708b	0.7416a
6	11/20	0.7416b	0.6708a
7	13/20	0.8062b	0.5916a
8	15/20	0.8660b	0.5000a
9	17/20	0.9220b	0.3873a
10	19/20	0.9747b	0.2236a
		Mean =	0.6684

TABLE 4

No.	Area	y	x
1	1/10	0.3162b	0.9487a
2	3/10	0.5477b	0.8367a
3	5/10	0.7071b	0.7071a
4	7/10	0.8367b	0.5477a
5	9/10	0.9487b	0.3162a
		Mean =	0.6713a

$$\frac{1.6713}{1.6667} = 1.0028$$

Volume of true ellipsoid with cylinder $= 1.6667\pi\, a\, b^2$
Volume by above method $= 1.6684\pi\, a\, b^2$

The Equation of the Ellipse is

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1$$

31 It is seen that there is practically no error in calculating the mean velocity from a traverse with 20 points as indicated above. It will be noted that the center velocity is omitted in the calculation.

32 Next let us investigate the error when only ten points are used, as shown in Fig. 7. In this case the areas of the concentric circles will be 1/10, 3/10, 5/10, 7/10, and 9/10. The results are given in Table 4. The error in this case is only 0.28 of 1 per cent, which is well within the error of observation. In this case also the center velocity is not used.

33 The Pitot tube used by the writers in this investigation, is not of the form that they would recommend for general use in cases of distorted flow, although it is well adapted to traversing straight pipes where a wall piezometer gives correct mean pressure indications. It was the only available tube suitable for the pipe. In general where a wall piezometer is not available, a tube with both impact and static openings would be more convenient, less liable to errors of observation and would effect a material saving of time.

34 The method above described, of obtaining the true velocity, at any point by means of the tube used corresponds to working with a

tube having both openings. In the latter case it is, of course, unnecessary to reverse the tube. The corresponding suction, if there be any, on the static openings may be obtained by using a wall piezometer in rating the tube.[1] Having rated the tube, either with the wall piezometer or by tank measurement of the discharge, the wall piezometer is no longer necessary to give pressure indications. A tube may be designed so that there will be no suction action at the static openings, i. e., the static openings will act precisely like wall piezometers and will give the true pressure. In this case the coefficient of the tube is unity.

35 In case the coefficient of the tube is not unity, and the impact point is of proper shape, the following statement gives the method of obtaining the suction head at the static openings.

36 In general we will have for the formula to be used with any Pitot tube,

$$v = c\sqrt{2gH}$$

where

$$H = (h + h_1),$$

That is, H is the water column difference between the impact and static openings.[2] Then

$$v^2 = c^2\, 2\, g\, H \text{ or } H = \frac{v^2}{c^2\, 2\, g}$$

37 The impact opening of the tube gives the head

$$h = \frac{v^2}{2g}$$

The static openings give the head

$$h_1 = (H - h) = \left(\frac{1}{c^2} - 1\right)\frac{v^2}{2g}$$

38 Thus, if the rating of a two-opening tube shows a coefficient of 0.883, we have

$$h_1 = \left(\frac{1}{0.883^2} - 1\right)\frac{v^2}{2g} = (1.284 - 1)\frac{v^2}{2g} = 0.284\frac{v^2}{2g}$$

[1] Water measurements in connection with a Test of a Centrifugal pump at Tourdan avenue Drainage Station, New Orleans, La,

[2] See article by W. M. White, Journal of Association of Engineering Societies, August 1901, p. 64.

CONCLUSION

39 To sum up:

a A method has been given for determining, by means of a Pitot tube with a single opening, the variation of velocities across a curved passage in which water is flowing.

b The variation of pressures across a curved passage has been determined by means of a Pitot tube with a single opening. So far as the writers know, this method has never been suggested before.

c A method has been pointed out by which, with a tube having both impact and static openings, the pressure at any point across a curved passage may be determined.

d By inference, the Pitot tube offers a solution of problems in the perfecting of the designs of turbines, pumps, etc., where it is important to know precisely how the fluid flows through the curved passages, especially how the velocities and pressures vary.

THE HORSE POWER, FRICTION LOSSES AND EFFICIENCIES OF GAS AND OIL ENGINES

By LIONEL S. MARKS, CAMBRIDGE, MASS.
Member of the Society

For a whole century the indicated horse power of the steam engine has been accepted as the most satisfactory measure of the work done in the cylinder of that engine. When the gas engine came into use it was but natural that the same measure of its power should be used. So long as the whole cycle took place in one cylinder there was but little doubt as to what was meant by the indicated horse power of the engine; but when auxiliary air and gas pumps were used, the indicated horse power required special definition. A committee of this Society reported in 1902 a code of rules which contained this special definition, and which has given a consistent meaning to the indicated horse power of steam engines and of gas and oil engines of all types.

2 The definition referred to is not, however, universally accepted. The pages of the *Zeitschrift des Vereins deutscher Ingenieure* for 1905 contain a long and most animated discussion by many of the ablest German engineers on the meaning of indicated horse power and mechanical efficiency in two-cycle engines, and they show very marked differences of opinion as to the correct method of calculating those quantities.[1] Within the past few months the definition of the indicated horse power of a four-cycle engine has been the subject of debate by the British Institution of Mechanical Engineers, and a strong tendency manifested itself to take as the measure of the indicated work of a four-cycle engine only that area which is included in the positive loop of the indicator card.

To be presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

The professional papers contained in Proceedings are published prior to the meetings at which they are to be presented, in order to afford members an opportunity to prepare any discussion which they may wish to present. They are issued to the members in confidence, and with the understanding that they are not to be published even in abstract, until after they have been presented at a meeting. All papers are subject to revision.

The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions. C55.

3 In all cases it has been assumed that the indicated horse power is the best measure of the work done in the engine, but the differences of opinion as to the methods of its measurement are really indications of the fact that the indicated horse power of an engine, and the quantities deduced from it, do not give that information which engineers have been trying to extract from them. The fundamental trouble with the indicated horse power as the unit of power is that it does not represent the actual work done by the working substance, but the difference between that quantity and certain resistance. Consequently it does not permit a comparison to be made between the actual amounts of work done by the working substances in the cylinders of engines of different types.

4 So far as the steam engine is concerned, the indicated horse power is certainly the most convenient and probably the most prac-

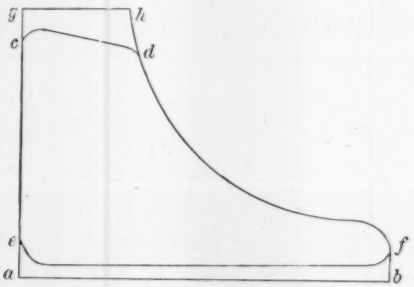

FIG. 1 INDICATOR CARD OF STEAM ENGINE

tical method of stating the amount of work that is done in the engine; but *for gas and oil engines, it is possible to use another method of stating the power of the engine; a method which is not only more convenient and practical but which also gives more information as to the real actions taking place, and forms a better basis for the comparison of the thermodynamic performance of engines of different types.*

5 The indicated horse power of a steam engine is really the algebraic sum of two quantities; these are (*a*) the total work done by the steam *inside the cylinder* and (*b*) the negative work done in overcoming the frictional and inertia resistances of the steam during the exhaust period. Thus in Fig. 1, if *a b* represents the pressure in the space into which the steam is exhausted, the total work done by the steam *in the cylinder* is *a c d f b*, and of this the amount *f e a b* is used up in over-

coming the resistances to the escape of the steam, leaving the area *e c d f* as the indicated work, or the total work done on the piston.

6 The work done in the cylinder is not however the total work done by the steam, since the steam has to do work in order to overcome the resistances to its admission. In Fig. 1, if *g h* represents the boiler pressure, the area *g h d c* is the work that the steam has to do in order to flow from the boiler to the cylinder. The total work done by the steam is consequently the area *g h b a*; and the area *c d f e* (i. e., the indicated work) is the difference between the total work done by the steam and the work necessary to get the steam into and out of the cylinder. The total work done by the steam in a steam engine is not shown directly on the indicator card, but has to be obtained by

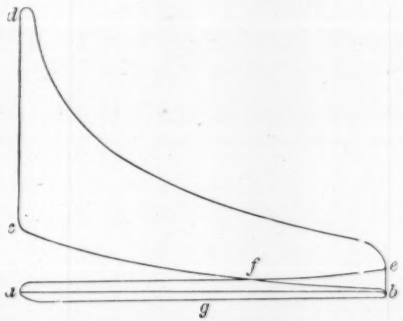

FIG. 2 INDICATOR CARD OF FOUR-CYCLE GAS ENGINE

drawing in the boiler pressure line and prolonging the indicator card to meet it.

7 The indicated horse power of gas and oil engines is defined by the Code of 1902 of the Committee on Standardizing Engine Tests, as the power developed in the engine cylinder (the algebraic sum of positive and negative works) minus the power indicated in the separate compression or feed cylinders, if there are any.

8 In a four-cycle engine, according to this definition, the indicated work is equal to the difference between the areas *c d e f* and *f a g b*, Fig. 2. If the area *e f b* be added to each of these, the indicated work is seen to be the difference between the areas *c d e b* and *e a g b*. The area *c d e b* is the total work done by gas; the area *a g b* is the work done in overcoming the resistances to the admission of gas, and the area *e a b* is the work done in overcoming the resistances to the exhaust of the gas; or in other words the area *e a g b* represents the

work that has to be done to get the charge into and out of the cylinder. The indicated work of a four-cycle engine is consequently seen to be the difference between (1) the total work done by the gas, and (2) the work necessary to get the gas into and out of the cylinder.

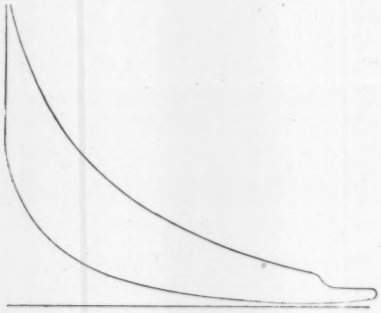

FIG. 3 INDICATOR CARD OF TWO-CYCLE GAS ENGINE

9 In a two-cycle engine with separate air and gas pumps, or with preliminary compression of the charge in the crank case, the indicated horse power, according to the definition, is the difference between (1)

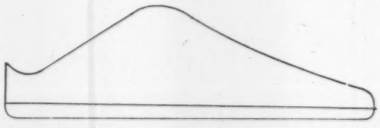

FIG. 4 INDICATOR CARD OF AIR PUMP

the main cylinder horse power, Fig. 3, and (2) the indicated horse powers of the air and gas pumps, Fig. 4 and 5, or of the crank case, Fig. 6. In this engine the exhaust occurs only near the end of the stroke, so that

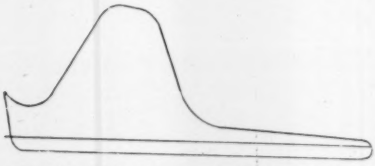

FIG. 5 INDICATOR CARD OF GAS PUMP

the amount of work done by the main cylinder piston in overcoming the resistance to the escape of the gases is so small as to be practically negligible. The work represented by Fig. 3 is the total work done by the gas; while the work represented by Fig. 4, 5 and 6 is the work done in overcoming the resistance to the admission of the charge and

consequently, in part, the work done in overcoming the resistance to the exhaust, since the incoming charge helps to force out the exhaust gases. The indicated horse power of a two-cycle engine is seen to have practically the same meaning as the indicated horse power of a four-cycle engine.

10 In a Diesel motor, the conditions are the same as in an ordinary four-cycle engine, with the addition that work is done in com-

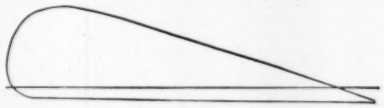

FIG. 6 CRANK CASE INDICATOR CARD

pressing the air used to spray the fuel. The indicated horse power of the air compressor must, according to the definition, be subtracted from the indicated horse power of the main cylinder in order to obtain the indicated horse power of the engine. The difference between the areas *c d e b* and *e a g b*, Fig. 7, is the indicated work of the main cylinder, and, as with the four-cycle engine, Fig. 2, it is the difference between the work done by the gas in the cylinder and the negative work

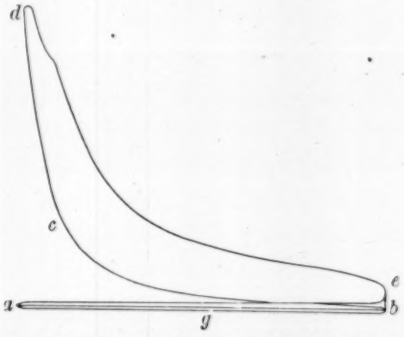

FIG. 7 INDICATOR CARD OF DIESEL ENGINE

done in overcoming the admission and exhaust resistances. The compressor card, however, Fig. 8, is different from the compressor cards for the two-cycle engine, Fig. 4 and 5, for it represents not only the work required to overcome the frictional resistances of admission of the fuel spray to the cylinder, but also the work of compressing the air used for spraying up to the pressure existing in the cylinder during the admission of the charge. It is obvious that if a large percent-

age of the air used per cycle in a Diesel motor were compressed in the air compressor instead of in the main cylinder, there would be a serious error in regarding the work done in the compressor as part of the frictional resistance to the admission of the fuel. In actual engines, the indicated work of the compressor pump is generally at least 6 per cent of the indicated work of the main cylinder. It is easily possible by drawing the cylinder admission pressure line *c d* on Fig. 8 to separate the work done there into its two components; the work done in compressing the charge (area *b*) and the work done in overcoming discharge resistances (area *a*). The frictional resistance to the admission of air to the compressor is too small to be shown on the diagram. The total work done by the charge is then the algebraic sum of the positive area *c d e b*, Fig. 7, and the negative area *b* Fig. 8, the work done in overcoming frictional resistances is the sum of the areas

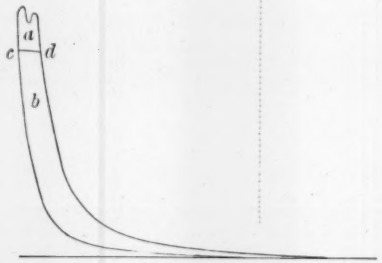

FIG. 8 INDICATOR CARD OF AIR COMPRESSOR DIESEL ENGINE

e a g b, Fig. 7, and *a*, Fig. 8. The indicated horse power of a Diesel motor has the same meaning as the indicated horse power of a four-cycle or a two-cycle engine; it is the difference between (1) the total work done by gas, and (2) the frictional resistances to the admission and exhaust of the gases.

11 In the analysis of the performance of a heat engine, there are two principal quantities that the engineer wants to know, namely, (*a*) the thermodynamic efficiency of the engine, or the percentage of the total heat going to the engine that is actually converted into work, and (*b*) the net efficiency of the engine, or the percentage of the total heat going to the engine that is available for doing useful work. The difference between these two efficiencies is the percentage of the total heat going to the engine that has been used up in overcoming the various resistances which the engine itself offers to the carrying out of the cycle of operations.

12 It has been the practice in the past to calculate the thermodynamic efficiency by finding the ratio of the indicated work to the total heat supplied. But this does not really measure the percentage of the total heat that has been converted into work; it measures the percentage of the total heat that is available for doing work after certain engine resistances, viz: those offered to the admission and exhaust of the working substances, have been overcome. The thermodynamic efficiency of an engine should have but one meaning and that is *the efficiency of the engine in converting heat into work*, irrespective of whether that work is used up, in part, in overcoming engine resistances or remains entirely available for useful applications. That is the plain meaning of the term and the only meaning which will permit a direct comparison of the efficiencies of the processes actually occurring in the cylinders of different engines. If the indicated work is used in calculating the thermodynamic efficiencies, such a comparison does not necessarily throw any light on the actual processes at all, since the frictional resistances resulting from a poor design of compression pumps, valves, ports, etc., may more than offset the gain from the use of a more efficient cycle.

13 It is, moreover, important that the thermodynamic efficiency should have the suggested meaning in order to permit a fair comparison with the ideal cycle. To state that the thermodynamic efficiency of a gas engine is 60 per cent of the thermodynamic efficiency of the ideal gas engine working under the same external conditions, is entirely misleading, if the gas friction resistances to admission and exhaust have been subtracted from the total work done by the gas. In some engines, the gas friction resistances may amount to 15 or 20 per cent of the total work, and if that were the case, an actual ratio of efficiencies of 70 per cent would appear to be but 60 per cent, that is, the apparent possibility of improvement of the purely thermodynamic processes would be reduced from 40 per cent to 30 per cent. If the gas friction is taken into account in calculating thermodynamic efficiencies, there does not seem any sufficient reason why the machine friction should not similarly be taken into account. The process of getting the charge into and out of the cylinder is purely mechanical—it is not part of the thermodynamic cycle.

14 The writer believes that for gas and oil engines, the power of the engine can be most usefully expressed as the total work done by the working substance—this might be called the *total horse power*, or, since it measures the amount of heat converted into work, the *thermodynamic horse power*. The total work for a four-cycle engine is the

area *cdeb*, Fig. 2; for a two-cycle engine, the area of Fig. 3; and for a Diesel engine, the area *cdeb*, Fig. 7, minus the work represented by the area *b*, Fig. 8, of the air compressor card. As measured in this manner, the total work is not entirely independent of the design of exhaust valves and passages since the occurrence of release before the end of the stroke (which is necessitated by the resistances of the exhaust) reduces the total work area. It is only in the case of the comparatively early exhaust of the two-cycle engine that the actual work might be considered as being affected in an appreciable manner by the release before the end of the stroke. It is, however, proper to regard the work of this cycle as being finished when the exhaust opens—the toe of the diagram being the equivalent of the negative area of the four-cycle diagram. Since the area of the toe of the diagram is always extremely small, its inclusion in the total work area introduces no appreciable error.

15 The total work done by the working substance is used up in three ways.

a In overcoming the resistances to the admission and exhaust of the charge; this may be called *gas friction work*.

b In overcoming engine friction; this may be called *machine friction work*, and

c In doing *useful work*.

16 The indicated horse power is then the total horse power minus the gas friction horse power and it retains the meaning it has always had.

17 An ordinary gas engine test permits the determination of the total horse power, the gas friction horse power, the machine friction horse power and the useful or brake horse power. The value of finding these separate horse powers will be apparent if, for example, a comparison is to be made between two-cycle and four-cycle gas engines. It is urged against the two-cycle engine that it obtains its very great advantage of nearly doubled power per cubic foot of piston displacement, at a cost of considerable loss in efficiency. This loss in efficiency is said to be (1) thermodynamic, resulting from (*a*) the loss of some of the charge to the exhaust during admission, or (*b*) the retaining of too much of the burnt gases in the cylinders; (2) gas friction loss resulting from the separate compression of the gas and air and the consequent extra valve and pipe resistance, and (3) machine friction losses resulting from the actual mechanical arrangements. The statement of the separate horse powers will throw light at once upon all

these points, and will show also wherein any particular engine fails to come up to the standard of its class.

18 From the commercial point of view, there is no advantage in retaining the indicated horse power, since it is the brake horse power that the engine user wants. From the scientific point of view, the indicated horse power can be of use only for the comparative study of engines and if it is not the best measure of power for that purpose, it should not be permitted to retain its present position.

19 If the total horse power, gas friction horse power, machine friction horse power and brake horse power are used as the standard measures of the engine power and losses, the various engine efficiencies could be defined in the following manner:

$$\frac{\text{Total horse power}}{\text{Total heat supply}} = \text{thermodynamic efficiency}$$

$$\frac{\text{Brake horse power}}{\text{Total horse power}} = \text{engine efficiency}$$

$$\frac{\text{Brake horse power}}{\text{Total horse power—Gas friction horse power}} = \frac{\text{Brake horse power}}{\text{Indicated horse power}}$$

$$= \text{Mechanical efficiency}$$

$$\frac{\text{Brake horse power}}{\text{Total heat supply}} = \text{net efficiency}$$

$$\text{Thermodynamic efficiency} \times \text{engine efficiency} = \text{net efficiency.}$$

20 These definitions retain for indicated horse power and mechanical efficiency their usual meanings.

21 The thermodynamic efficiency is the actual efficiency of the process of converting heat into work; the engine efficiency is the true measure of all the frictional losses of the actual mechanism, not only the friction of bearings and pistons, but also of the gas entering and leaving the cylinder; and the net efficiency is the quantity that interests the person who pays the bills for fuel. The mechanical efficiency has its use in showing the extent of machine friction losses, but unless the engine efficiency is also stated, it tends to obscure the real magnitude of the more or less avoidable friction losses in an engine.

22 There would be certain incidental advantages from the use of total horse power as the unit of measurement apart from the more important scientific advantages of a unit which means a single definite thing—and not the sum of two quantities of very different kinds. In ordinary practice, there is more complexity and greater possibility of inaccuracy in the measurement of the indicated horse power of gas and oil engines than is the case with steam engines. The greater complexity arises from the fact that it is necessary in the two-cycle engine to take indicator cards not only from the main cylinder but also from the auxiliary gas and air pumps or from the crank case, and for a Diesel engine, it is necessary to take cards from the air compressor as well as the main cylinder. The greater inaccuracy results from the fact that in going round the *negative* area of the four-cycle, or Diesel-cycle cards, the probable planimeter error has the same absolute magnitude as in going round the *positive* area, and these two errors may both be of the same sign. If, to avoid this, a weak spring diagram is taken of the work of the exhaust and suction strokes, we have the complexity of another indicator. Of course when scientific results are needed, in which case the gas friction horse power must be obtained, it will be necessary to take cards from all the auxiliary cylinders, and the greater complexity cannot be avoided; but for ordinary commercial purposes, if any measurement of power is required beside the brake horse power, the total horse power would serve quite as well as the indicated horse power, and it could be obtained more easily and with more accuracy. Commercially, the indicated horse power is of no particular use when the brake horse power is known, and scientifically it is less useful than the total horse power.

23 The proposed new measure of power cannot be conveniently applied to the steam engine, nor does it seem desirable to so apply it, since the practice in that case is firmly fixed. In the steam engine, part of the compression work is carried out in the air and feed pumps, but the indicated work in these auxiliaries is not taken into account in calculating the indicated horse power: i. e., a different practice exists from that which this Society recommends as proper for the determination of the indicated horse power for gas and oil engines. The history of the steam engine is probably more in the past than in the future, so that a change in the practice is not particularly desirable even if practicable: but the history of the gas and oil engines is almost entirely in the future, and a proper choice of the units of power may help to make that history more clear.

24 In conclusion, the writer wishes to submit to the Society the desirability of an early revision of the code of rules for carrying out and reporting gas and oil engine tests. The remarkable extension in the use of the gas engine, the growth of the large variety of types which that has stimulated, the considerable body of research throwing light on that motor which has been published since the appearance of the code, have made it apparent that the code is deficient in certain respects, and have rendered it desirable that many changes should be made.

25 When such revision is made, the writer hopes that there may be incorporated in it the suggestions as to horse power and efficiencies which he has presented above.

THE THERMAL PROPERTIES OF SUPERHEATED STEAM

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Member of the Society

The purpose of this paper is to compare, discuss and combine the best available data in regard to the specific heat of superheated steam, to determine as nearly as may be the true value and manner of variation of that quantity, and to derive a numerical table which shall be as reliable and convenient for general use as is the ordinary table of the properties of saturated steam.

2 The discussion will be based almost altogether upon the experimental researches of Knoblauch and Jakob and of Thomas. The former were published in the *Zeitschrift des Vereins deutscher Ingenieure*, in January 1907, at pp. 81 and 124; the experiments were made in the laboratory of technical physics at the Royal Technical High School, Munich, in the latter part of the year 1905; and the results were brought before this Society by Professor Greene at the Indianapolis meeting, in 1907. The paper of Professor Thomas, setting forth experiments made at Sibley College, Cornell University, was presented to the Society at the Annual meeting, 1907, and will be found in the Proceedings for December 1907.

3 In each paper is given a set of curves for the specific heat at constant pressure, c_p, the individual curve showing how c_p varies with the temperature during the operation of superheating at some particular pressure. To superimpose the two groups of curves on one plane would be very confusing; they can be much more effectively compared by means of the "solid" or three dimensioned diagram drawn in Fig. 1.

To be presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

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4 As indicated, one abscissa of the base plane is the temperature t in degrees fahrenheit, parallel to the axis OT, the other is the absolute pressure p in pounds per square inch, parallel to OP, while c_p is the

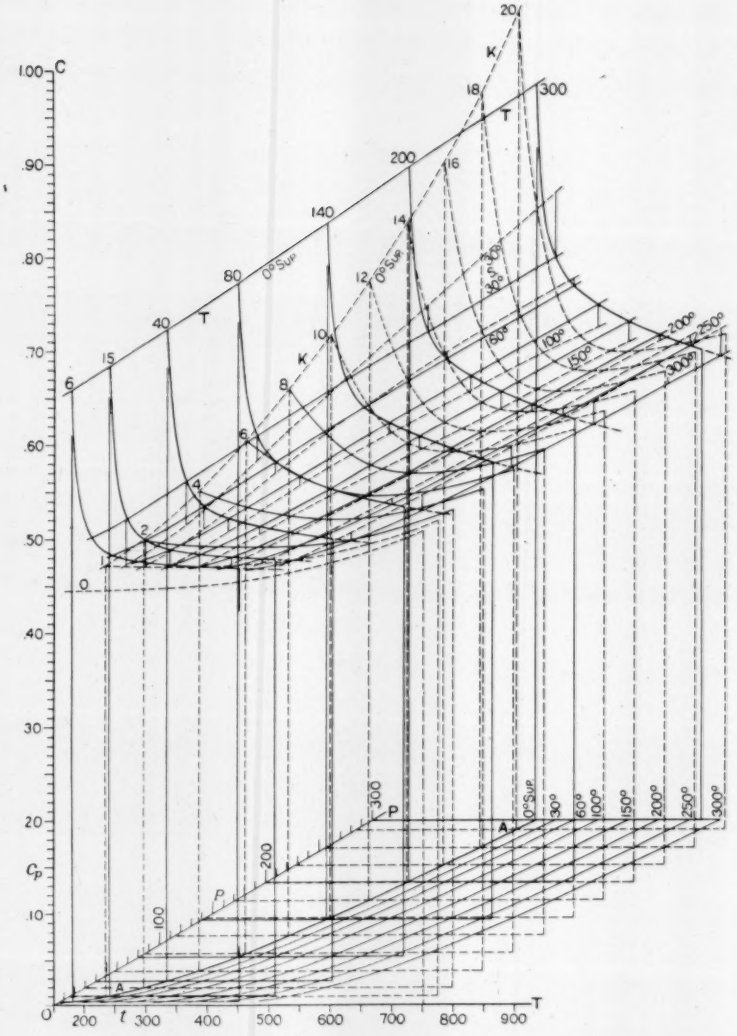

FIG. 1. COMPARISON OF CURVES OF SPECIFIC HEAT

vertical ordinate. The primary curves, each on its own TC plane, are redrawn directly from Fig. 6 of Thomas' paper and from Fig. 11 of Knoblauch and Jakob. The latter figure is reproduced as Fig. 1 in

Greene's paper, in Proceedings for June 1907. The law of variation of c_p with temperature and pressure, as determined by each experimenter, is now represented by a curved surface; and by drawing cross curves we have an easy means of interpolating so as to get TC curves in both systems at the same pressures (see Fig. 4), or can use the cross curves directly for comparison.

5 The line joining the high initial points of the TC curves, at the left, is the line of saturation or of zero superheat; it lies on a vertical curved surface which cuts the base plane in the curve AA, the latter being the fundamental curve of relation between the temperature and the pressure of saturated steam. Along the saturation lines the letters K and T designate the respective results of Knoblauch and Jakob and of Thomas, and these letters will be used as abbreviations hereafter. The numbers on the T curves show pressures in pounds, while those on the K curves show kilograms per square centimeter, the relation being, 1 kg. per square centimeter = 14.22 lb. per square inch. The T curves are all in full line, except the dotted extensions beyond the limit of actual determination at 270 deg. of superheat. In general, the K curves are dotted, except the four which show actual experiment, namely, those for 2, 4, 6 and 8 kg. These curves are stopped at 400 deg. cent., or at 752 deg. fahr., although the determinations really extended only to about 700 deg. fahr., as is made clear on Fig. 4.

6 The cross curves are drawn for constant superheat. On the base plane PT a number of curves are put in at a constant distance in the temperature direction, from the original pressure-temperature curve AA—or, in effect, this curve is shifted various distances to the right. From the intersections with the base lines of the TC curves, vertical lines are drawn to the curves themselves; or, more simply, measurements of 30 deg., 60 deg., 100 deg., etc., are made directly from the lefthand limiting ordinates to locate points on the curves. Through these points the cross curves are traced, and marked with the degree of superheat; and further, short verticals are extended from every intersection point on the T surface to the corresponding cross curve on the K surface, in order to show how far apart the two surfaces are, or to get points on a primary K curve at the same pressure as each T curve.

7 Examining Fig. 1, and bearing in mind the fact that the K curves beyond 8 kg. are got by extrapolation, we note that from 50 or 60 deg. of superheat out to the limit of the experiments the two sets of results agree very well; but that there is a marked difference near

saturation and the beginning of a wide divergence toward the region of higher superheat. We shall now take up the two questions suggested by these discrepancies.

SPECIFIC HEAT AT AND NEAR SATURATION

8 Near to the boundary line which separates the superheated from the saturated state, it appears that there are some residual molecular attractions to be overcome, and for this reason the specific

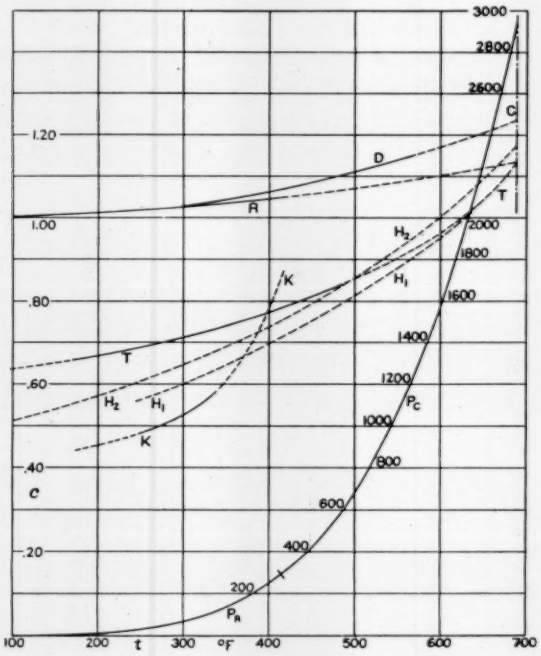

FIG. 2 CONDITIONS AT THE SATURATION LIMITS

heat is higher at and near this lower limit of superheat than it is farther out. A number of considerations affecting the conditions just at this limit are set forth graphically on Fig. 2, where the base line is temperature fahrenheit. The four curves marked T, K, H_1 and H_2 show the specific heat c_p of saturated steam along the saturation line, under various conditions or assumptions; the curves R and D give the specific heat of water, which rises well above unity with increasing temperature; the curve designated by the letter P shows the relation between pressure and temperature for saturated steam.

The lower part of the last curve, marked P_R and extending to the cross-line at about 300 lb. pressure, gives Regnault's determination; the upper part P_C is from the experiments of Cailletet and Colardeau, published in Annales de Chemie et de Physique, sixth series, vol. 25, 1892, p. 527. It runs up to the critical temperature as determined by these investigators, which is at 689 deg. fahr. and at a pressure of 2946 lb. per square inch. The numbers marked along this curve give the scale of pressure.

9 Of the two curves for the specific heat of water, the one marked R is from Regnault's formula,

$$c = 1.0000 + 0.000\,00222\ (t - 32) = 0.000\,000\,2778\ (t - 32)^2,$$

which is based on experiments extending to about 425 deg. fahr. The curve D is from Dieterici's experiments, published in the *Zeitschrift des Vereins deutscher Ingenieure*, March 1905, p. 362, as represented by the formula:

$$c = 0.9983 - 0.000\,0576\ (t - 32) + 0.000\,000\,64\ (t - 32)^2$$

Here the limit of actual determination was 300 deg. cent. or 572 deg. fahr., and this formula is to be accepted for the higher range of temperature.

10 The reason for presenting these apparently irrelevant data will now be made apparent. The specific heat curves K and T, for c_p at zero superheat, laid out with ordinates transferred to Fig. 4 directly from Fig. 1, and with the range of experimental determination shown in full line for each case, represent radically different ideas as to the manner of variation of this quantity. Knoblauch and Jakob derive a formula for the curve which passes through the inner ends of their four C T curves, putting this formula into such a shape that it makes c_p equal infinity at the critical temperature of 365 deg. cent. or 689 deg. fahr., and remarking casually that the mechanical theory of heat calls for this value at that point.[1] The curve taken from Thomas' presentation conforms very well to the idea that since hot water and superheated steam merge into a common state at or above the critical temperature, the two specific heats, coming up along and just outside of the inner and outer limits of the condition of wet or saturated steam, ought to run into the same value, decidedly a finite number, somewhere near the critical point. That the latter is, in a general way, the correct view is the opinion of the writer, who further

[1] Zeuner, in his Technical Thermodynamics, at the end of art. 38, p. 312 of vol. 2 of the recent translation by Professor Klein, makes the same casual statement. The writer does not know who first developed the idea.

believes that the idea of infinite c_p at the critical temperature is based on a misconception.

THE CRITICAL STATE OF WATER

11 In order to clear up the difference of opinion or of conception suggested in the last paragraph, and to get a reasonable indication of the trend of the initial c_p toward the higher ranges of pressure, we must consider more fully what is involved in the term "critical state" or "critical point" of a fluid. In one definition, it is a limiting temperature, above which the substance cannot exist in the liquid state; according to another, it is a condition where the two states of liquid and of superheated vapor merge into each other, the ordinary operation of evaporation into saturated vapor disappearing. Under the latter idea, the latent heat of vaporization, which decreases with the temperature of steam formation, will become zero at the critical point. Not to cite other writers, we may refer to the paper by Mr. C. V. Kerr on "The Potential Efficiency of Prime Movers," presented to the Society in 1904, and published in the Transactions, vol. 25, p. 920. It is there stated that by producing the two limit curves on the entropy-temperature diagram until they met, the point where the latent heat vanishes was found to be at a height of 975 deg. fahr. This method is only the roughest kind of an approximation; but when we compare the result with the value 689 deg. quoted above, there appears to be a wide discrepancy.

12 The experiments of Cailletet and Colardeau consisted in observing the simultaneous pressure and temperature of a small mass of water in a closed metal tube. They found that the single, definite relation between p and t, characteristic of all ordinary ranges, ceases to exist at 365 deg. cent., or 689 deg. fahr. We shall now proceed to show that this "critical point" is by no means the same as that defined above.

13 In Fig. 3 is drawn the usual pressure-volume diagram for one pound of water substance, but with a very small scale of pressures and a very large scale of volumes: $A\,B$ is the line of water volume, or the inner limit of saturation, between water and wet steam; $C\,D$ is the dry steam curve or the outer limit of saturation, separating wet steam from superheated. The horizontal line $B\,C$ is drawn at the pressure corresponding to 689 deg. fahr., or at 2946 lb. The limit curves $A\,B$ and $D\,C$ are laid out according to determinations made at comparatively low pressures, and are therefore merely tentative; but they

decidedly do not meet at the height $B\,C$. We must believe rather that $B\,C$, as determined by coming up from below, is one limit of a quite widely extended "critical state," of which the upper limit is at some point like E which is located by the convergence of curves produced from $A\,B$ and $D\,C$.

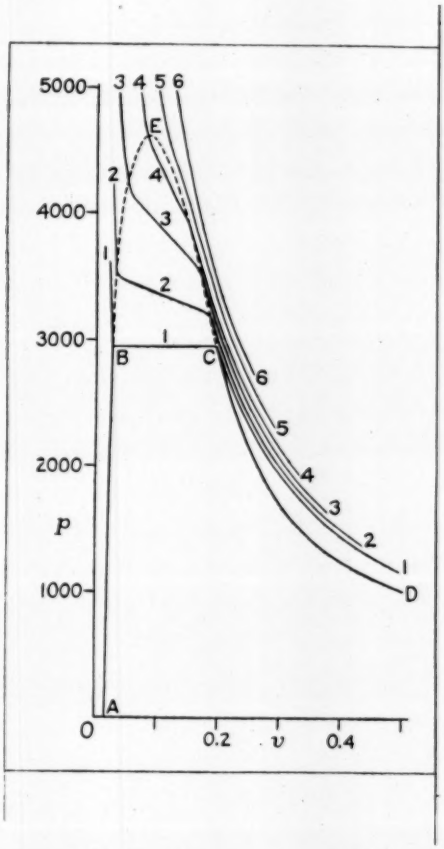

FIG. 3 THE CRITICAL STATE OF WATER

14 The most distinctive characteristic of the ordinary mixture of steam and water, thermodynamically considered, is found in the fact that the operation of evaporation at constant pressure takes place isothermally. For a so-called perfect gas the isothermal operation follows the law $p\,v = \text{const.}$, and this is very nearly realized with any actual dry gas. The complete isothermal of steam, at any moderate temperature, consists of three parts; viewing it as a compression,

we have a curve approximately of the form $p\,v = C$ in the superheated range, a horizontal line of condensation across the range of wet steam, and a nearly vertical line for the very slight compression of water by increasing pressure. At each limit line, $A\,B$ and $C\,D$ of Fig. 3, there is an abrupt change of characteristic.

15 Now the critical point as found by Cailletet and Colardeau is the upper limit of a condition which exists throughout the lower ranges of temperature. It seems reasonable to assume that it is also the upper limit of the other condition just described, or that the complete isothermal numbered 1 in Fig. 3 is the last one that has a horizontal or constant pressure portion. Above $B\,C$ the isothermals must begin to slant upward in crossing the space between the limits marked by extending $D\,C$ and $A\,B$; only by some such change can they gradually approach the simple continuous curve which must exist for very high temperatures. The changes at the limits $B\,E$ and $C\,E$ probably become less abrupt, or less strictly localized, as the temperature is higher, but there must be some such locus of these changes as is here traced in the outline $B\,E\,C$.

16 The condition $c_p = \infty$, which means that heat can be imparted to a substance at constant pressure without raising the temperature, is characteristic of the steam and water mixture whose mechanical condition is represented by any point within the enclosure $A\,B\,C\,D$. Above $B\,C$, c_p has a finite but gradually decreasing value, since a change of state at constant pressure across the area $B\,E\,C$ from left to right involves an increase in temperature. To the left of $B\,E$ the substance is liquid, or is in a state of close conglomeration which is analogous to the definite liquid condition farther down; and along $B\,E$ the specific heat will probably continue to increase somewhat after the manner of curve D on Fig. 2. The last (highest) isothermal that just touches the region $C\,E\,B$ will be tangent somewhere below E, and it seems reasonable that the highest value of the specific heat just along the outside of $C\,E\,B$ should be at about this highest temperature.

17 The idea that $c_p = \infty$ at the critical point is based on the assumption that the isothermals across $B\,E\,C$ are horizontal clear up to the vanishing point; then at the rounded apex E we should have an instantaneous condition of constant pressure and constant temperature, which would satisfy the physical meaning of the mathematical expression. The discussion just completed shows that the highest temperature along the curve $B\,E\,C$ is probably below E, toward C, and that in any case, the point E will be far above the limit of the true liquid state at B.

CONCLUSIONS AS TO INITIAL SPECIFIC HEAT

18 Without indulging in further speculation as to the behavior of water above the limit marked by the line BC of Fig. 3, we shall now resume consideration of the results of Knoblauch and Jakob, and take up the question whether they really are correctly interpreted, close to saturation, by the curves drawn by the authors. The method of these experiments, briefly stated, was as follows: The current of steam was first superheated to a desired degree in a preliminary electrical heater, then further raised in temperature by a measured supply of electrical energy in a second heater which really constituted the determining apparatus of the experiment. The result obtained

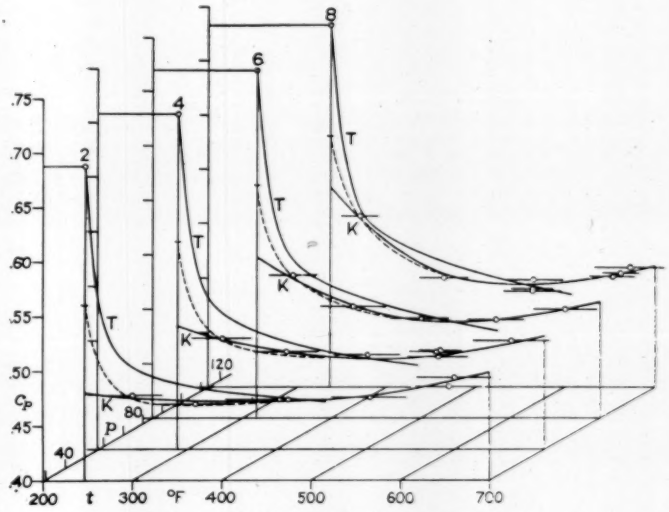

FIG. 4 CLOSE COMPARISON OF CURVES c_p

was the mean specific heat over the range from t_1 at entrance to t_2 at exit of the final heating coil. The published tabular results are replotted on Fig. 4, where each circled point shows a value of the mean specific heat, while the short horizontal line marks the range over which the determination extended. The four original curves, as published, are drawn in full line and marked K; with them are drawn also curves from Thomas' results, carefully interpolated on Fig. 1 for the pressures 2, 4, 6 and 8 kg. and marked T. The lowest determination by Knoblauch and Jakob began at from 12 to 18 deg. fahr. above saturation.

19 Now the dotted curves in Fig. 4, drawn by the writer, show how without putting the least strain upon the actual experimental results, the Knoblauch and Jakob curves of c_p can be made to take a form closely similar to that of Thomas' curves. The saturation line of c_p thus obtained is laid out in the curve marked H_1 on Fig. 2, where it easily runs into the extended T curve. The writer believes that this is the correct interpretation of the results of Knoblauch and Jakob; and with this interpretation the two sets of experiments strongly confirm each other.

20 Finally, taking what seems to be the most probable mean between the curves T and H_1 over the range of experiment, and further considering that if the curves of c_p and of c_w (the latter the D curve for water) are to meet somewhere beyond the critical ordinate C, the former must be lower at this ordinate because of its more rapid upward slant, the writer has laid out the curve H_2 on Fig. 2 as representing, in his judgment, the truest location that can be made, from present data, of the curve of c_p for zero superheat. This is used as the starting point in the final layout of c_p in Fig. 6 and 7.

THE HIGHER RANGE OF SUPERHEAT

21 As remarked in Par. 7, the two sets of curves show the beginning of a marked divergence toward the higher ranges of superheat. Most of the available data for these ranges have been obtained by the explosion method. This consists in enclosing a combustible mixture of gas (hydrogen) and air (or oxygen) in a vessel, preferably of spherical form, starting ignition at the center of the vessel so that a spherical flame-cap, propagated at a uniform rate, will reach the whole containing surface at the same time, and measuring the resulting instantaneous high pressure before there is a chance for the heat to be dissipated by radiation. This leads to the specific heat at constant volume, from which that at constant pressure can be derived by means of the fundamental physical relation between the two, $c_p - c_v = \text{const.}$ The results obtained, all expressed as rectilinear or first-degree functions of the temperature, exhibit rather a wide divergence; but Knoblauch and Jakob show, at the end of their paper, how the assumption that both the mean and the instantaneous values of c_p increase with t according to a curve relation, or at an increasing rate leads to a very close reconciliation of these conflicting indications. Further the curve which they draw agrees very well with the one set of experiments at constant pressure, carried out to quite a fairly high temperature, that is now available.

22 These experiments were made by Holborn and Henning at the Physical Technischen Reichsanstalt, and were published in Annalen der Physik, vol. 18, 1905, p. 739. Steam at the pressure of the atmosphere was electrically superheated, then passed into a small absorption calorimeter of the usual type, but with oil instead of water as the heat absorbent. In the four sets of experiments the upper temperature was 270 deg., 440 deg., 620 deg., and 820 deg., cent.; and in each case the steam was cooled to about 110 deg., leaving the calorimeter with this small residuum of superheat. The mean specific heat over the range of cooling was the quantity determined, and the four values are plotted on Fig. 5.

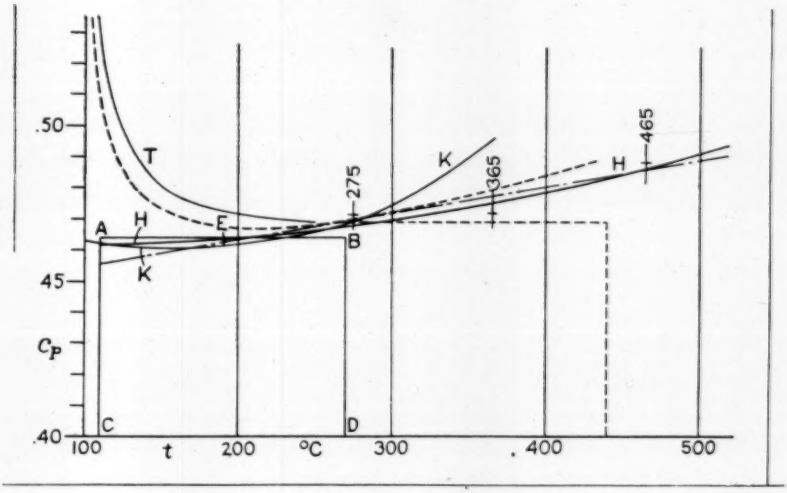

FIG. 5 SPECIFIC HEAT AT ATMOSPHERIC PRESSURE

23 The first experiment, from 270 deg. to 110 deg. is represented by the rectangle *C A B D*, the height of *A B* showing what the specific heat would be if it were constant over the whole range, and the middle point *E* locating the mean specific heat at the middle of the range. The three other middle parts, similar to *E*, are marked by short cross lines on the ordinates at 275 deg., 365 deg., and 465 deg., Now the curve of mean specific heat would go through the outer corners of a series of rectangles like *C A B D*, or through the *B* points; with rectilinear variation, the line of true specific heat would go through the middle points like *E*; and a curve of true specific heat should be of such form that its mean height over any given range such as *C D*, or the area beneath the curve, will be the same as that of the rectangle

like *CABD*. On Fig. 5 the straight dot and dash line *H* shows the Holborn-Henning formula; the curve running with this line has been traced by the writer as probably a truer interpretation of the results.

24 The practical application of this last discussion is made evident when we draw in the specific heat curves of Knoblauch and Jakob and of Thomas for the same (atmospheric) pressure, designating them by *K* and *T* as heretofore. It appears that the *K* curve rises much too rapidly, while the *T* curve fails to show the incipient upward tendency which it ought to have.

FINAL DETERMINING CURVES

25 Using this combination of data as a basis of judgment, and bearing in mind the modification of the *K* curve made on Fig. 4, the writer has traced in the dotted curve shown on Fig. 5 as representing his conclusion in regard to the most probable value of c_p at atmospheric pressure. Near saturation there is a considerable difference between the *T* curve and the other two, although the disagreement is exaggerated by the large vertical scale of the drawing. The difference exists in both the form of the curves and their absolute height. The *H* curve as drawn agrees very well with the original *K* curve, but there is no reason why a rise toward saturation may not be completely masked in the determination of the heat given off by the steam through a wide range of cooling. This *H* curve might easily, and with equally close conformity to the experimental data, be replaced by one of our final form, after the same method that was applied to the *K* curves on Fig. 4. The quantitative discrepancy is less easily overcome; neither set of experiments presents a predominating claim to confidence, and it appears that the best way to arrive at a conclusion is to "split the difference," noting at the same time that, at the most, this difference is not very great either absolutely or relatively.

26 The guiding curves used by the writer in finally laying out c_p on Fig. 7 are drawn on one plane in Fig. 6, for the pressures 15, 100, 300 and 600 lb. per square inch. These derived curves are in full line, marked *H*, while the corresponding *K* and *T* curves are dotted. At 15 lb. there is the discrepancy just discussed. At 100 lb. the agreement is much better, which is all the more satisfactory because this is within the range of ordinary technical application in the steam plant. At 300 lb. very little weight is given to the widely extrapo-

lated K curve. The H curve at 600 will give a much "fairer" surface in Fig. 7 than would the T curve.

RESULTS

27 In Fig. 7 and Table 1 are set forth the final results of the discussion. The diagram, which shows c_p, is laid off in the same general manner as Fig. 1 and 4, but the temperature abscissa is degrees of superheat t_s, instead of temperature on the thermometer scale. Then the cross curves, which give the variation of c_p with change in pressure, are in vertical planes instead of lying on vertical curved surfaces as in Fig. 1. The full line curves cover the whole range, from

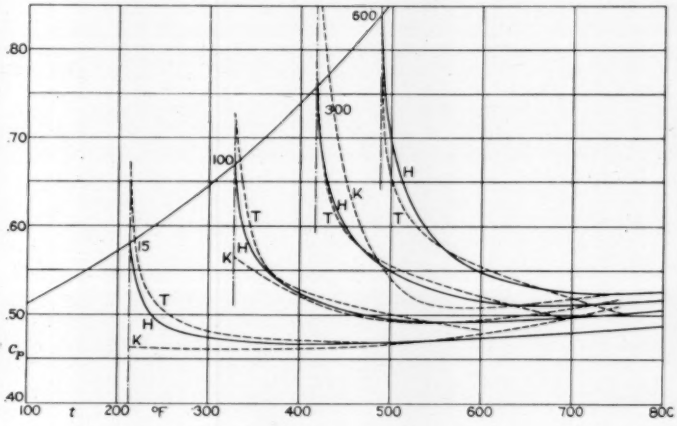

FIG. 6. DETERMINING CURVES

1 lb. to 600 lb. pressure; the dotted curves show the low-pressure range, up to 50 lb., spread out on a pressure scale ten times as large as that of the main figure. Values of c_p are measured from the base lines on the bottom plane PT to intersections of the curves on the c_p surface. Fig. 7, as here reproduced, is too small to have any but an illustrative value: measurements from the original full size diagram are given in Table 1.

28 In the table, the quadruple horizontal lines correspond with the TC curves on Fig. 7, giving quantities which belong to an operation of heating at the constant pressure marked at both outer margins: the columns are analogous to the PC cross curves on Fig. 7, showing variation with the pressure for a particular range of superheat as

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM

P		DEGREES OF SUPERHEAT							
		0	10	20	40	60	80	100	130
	t	102.0	112.0	122.0	142.0	162.0	182.0	202.0	232.0
1	c	.513	.475	.465	.457	.454	.453	.453	.453
	h	(1113.1)	4.9	9.6	18.8	27.9	36.9	46.0	59.6
	n	(1.9890)	.0086	.0167	.0323	.0472	.0615	.0754	.0955
	t	141.6	151.6	161.6	181.6	201.6	221.6	241.6	271.6
3	c	.535	.490	.474	.463	.459	.457	.456	.456
	h	(1125.1)	5.1	9.9	19.2	28.4	37.6	46.7	60.4
	n	(1.8891)	.0083	.0161	.0309	.0451	.0587	.0719	.0910
	t	170.1	180.1	190.1	210.1	230.1	250.1	270.1	300.1
6	c	.553	.502	.484	.471	.466	.463	.461	.460
	h	(1133.8)	5.2	10.1	19.7	29.0	38.3	47.6	61.4
	n	(1.8276)	.0082	.0158	.0303	.0441	.0573	.0702	.0887
	t	193.2	203.2	213.2	233.2	253.2	273.2	293.2	323.2
10	c	.567	.515	.494	.479	.473	.470	.467	.465
	h	(1140.9)	5.4	10.4	20.1	29.6	39.0	48.4	62.4
	n	(1.7832)	.0081	.0157	.0299	.0434	.0565	.0691	.0873
	t	213.0	223.0	233.0	253.0	273.0	293.0	313.0	343.0
15	c	.581	.526	.504	.487	.480	.475	.472	.469
	h	(1146.9)	5.5	10.6	20.5	30.2	39.7	49.2	63.3
	n	(1.7487)	.0081	.0156	.0296	.0430	.0559	.0683	.0862
	t	227.9	237.9	247.9	267.9	287.9	307.9	327.9	357.9
20	c	.591	.535	.513	.494	.486	.480	.476	.473
	h	(1151.5)	5.6	10.8	20.9	30.7	40.3	49.9	64.1
	n	(1.7244)	.0081	.0155	.0295	.0428	.0555	.0678	.0856
	t	240.0	250.0	260.0	280.0	300.0	320.0	340.0	370.0
25	c	.600	.543	.521	.500	.491	.484	.480	.476
	h	(1155.1)	5.7	11.0	21.2	31.1	40.8	50.5	64.8
	n	(1.7058)	.0080	.0155	.0294	.0426	.0553	.0675	.0851
	t	250.3	260.3	270.3	290.3	310.3	330.3	350.3	380.3
30	c	.608	.550	.527	.506	.496	.489	.484	.479
	h	(1158.3)	5.7	11.1	21.4	31.4	41.3	51.0	65.5
	n	(1.6908)	.0080	.0154	.0293	.0425	.0552	.0673	.0848
	t	267.1	277.1	287.1	307.1	327.1	347.1	367.1	397.1
40	c	.620	.560	.537	.516	.505	.479	.491	.485
	h	(1163.4)	5.9	11.3	21.8	32.0	42.0	51.9	66.6
	n	(1.6677)	.0080	.0154	.0292	.0424	.0549	.0670	.0844
	t	280.9	290.9	300.9	320.9	340.9	360.9	380.9	410.9
50	c	.631	.570	.546	.524	.512	.503	.497	.490
	h	(1167.6)	6.0	11.5	22.2	32.5	42.4	52.6	67.4
	n	(1.6497)	.0080	.0153	.0292	.0422	.0548	.0668	.0841
	t	292.5	302.5	312.5	332.5	352.5	372.5	392.5	422.5
60	c	.640	.577	.553	.531	.517	.508	.502	.494
	h	(1171.2)	6.0	11.7	22.5	32.9	43.2	53.3	68.2
	n	(1.6354)	.0080	.0153	.0291	.0421	.0546	.0666	.0838

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM—Continued

DEGREES OF SUPERHEAT										p
160	200	250	300	350	400	450	500	600		
262.0	302.0	352.0	402.0	452.0	502.0	552.0	602.0	702.0	t	
.453	.454	.455	.457	.460	.463	.466	.469	.478	c	1
73.2	91.3	114.0	136.8	159.8	182.8	206.1	229.4	276.8	h	
.1147	.1392	.1680	.1953	.2211	.2458	.2693	.2919	.3344	n	
301.6	341.6	391.6	441.6	491.6	541.6	591.6	641.6	741.6	t	
.455	.456	.457	.459	.462	.465	.468	.471	.479	c	3
74.1	92.3	115.1	138.0	161.0	184.2	207.5	231.0	278.5	h	
.1093	.1326	.1603	.1864	.2112	.2350	.2577	.2795	.3207	n	
330.1	370.1	420.1	470.1	520.1	570.1	620.1	670.1	770.1	t	
.459	.459	.460	.462	.465	.467	.470	.474	.482	c	6
75.1	93.5	116.5	139.5	162.7	186.0	209.4	233.1	280.8	h	
.1065	.1291	.1260	.1815	.2057	.2289	.2512	.2725	.3130	n	
353.2	393.2	443.2	493.2	543.2	593.2	643.2	693.2	793.2	t	
.464	.463	.463	.465	.468	.471	.474	.477	.485	c	10
76.3	94.9	118.0	141.2	164.5	188.0	211.6	235.4	283.5	h	
.1047	.1270	.1533	.1783	.2022	.2250	.2469	.2680	.3080	n	
373.0	413.0	463.0	513.0	563.0	613.0	663.0	713.0	813.0	t	
.468	.467	.468	.470	.472	.475	.478	.481	.489	c	15
77.4	96.1	119.4	142.9	166.4	190.1	213.9	237.9	286.4	h	
.1034	.1253	.1513	.1761	.1997	.2223	.2440	.2649	.3046	n	
387.9	427.9	477.9	527.9	577.9	627.9	677.9	727.9	827.9	t	
.471	.470	.471	.473	.475	.478	.481	.484	.492	c	20
78.3	97.1	120.6	144.2	167.9	191.7	215.7	239.9	288.7	h	
.1026	.1243	.1500	.1745	.1979	.2204	.2419	.2627	.3022	n	
400.0	440.0	490.0	540.0	590.0	640.0	690.0	740.0	840.0	t	
.473	.472	.474	.476	.478	.481	.484	.487	.495	c	25
79.1	98.0	121.6	145.4	169.2	193.2	217.3	241.6	290.7	h	
.1020	.1235	.1491	.1734	.1967	.2190	.2404	.2611	.3004	n	
410.3	450.3	500.3	550.3	600.3	650.3	700.3	750.3	850.3	t	
.476	.475	.476	.478	.481	.483	.486	.490	.498	c	30
79.8	98.8	122.6	146.5	170.4	194.5	218.8	243.2	292.5	h	
.1016	.1230	.1484	.1726	.1958	.2180	.2393	.2599	.2991	n	
427.1	467.1	517.1	567.1	617.1	667.1	717.1	767.1	867.1	t	
.481	.479	.480	.482	.485	.487	.490	.493	.502	c	40
81.1	100.3	124.2	148.3	172.3	196.7	221.2	245.8	295.5	h	
.1010	.1222	.1474	.1714	.1944	.2164	.2376	.2581	.2971	n	
440.9	480.9	530.9	580.9	630.9	680.9	730.9	780.9	880.9	t	
.486	.483	.483	.485	.488	.491	.494	.497	.505	c	50
82.1	101.4	125.6	149.8	174.1	198.6	223.2	247.9	298.0	h	
.1006	.1216	.1466	.1705	.1933	.2152	.2363	.2567	.2955	n	
452.5	492.5	542.5	592.5	642.5	692.5	752.5	792.5	892.5	t	
.489	.486	.486	.487	.490	.493	.496	.500	.507	c	60
82.9	102.4	126.7	151.0	175.5	200.1	224.8	249.7	300.0	h	
.1002	.1211	.1460	.1697	.1924	.2142	.2352	.2555	.2941	n	

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM—Continued

p		DEGREES OF SUPERHEAT							
		0	10	20	40	60	80	100	130
	t	311.8	321.8	331.8	351.8	371.8	391.8	411.8	441.8
80	c	.655	.589	.564	.540	.525	.515	.508	.500
	h	(1177.0)	6.2	11.9	22.9	33.6	44.0	54.2	69.3
	n	(1.6132)	.0080	.0153	.0290	.0419	.0543	.0662	.0832
	t	327.6	337.6	347.6	367.6	387.6	407.6	427.6	457.6
100	c	.669	.599	.573	.547	.531	.521	.513	.504
	h	(1181.9)	6.3	12.2	23.3	34.1	44.6	55.0	70.2
	n	(1.5963)	.0079	.0152	.0289	.0418	.0541	.0658	.0827
	t	347.1	357.1	367.1	387.1	407.1	427.1	447.1	477.1
130	c	.687	.610	.583	.555	.538	.527	.518	.508
	h	(1187.8)	6.4	12.4	23.8	34.7	45.4	55.8	71.3
	n	(1.5769)	.0079	.0152	.0288	.0416	.0537	.0654	.0821
	t	363.3	373.3	383.3	403.3	423.3	443.3	463.3	493.3
160	c	.703	.621	.593	.562	.544	.532	.523	.513
	h	(1192.7)	6.5	12.6	24.2	35.3	46.0	56.6	72.1
	n	(1.5620)	.0079	.0152	.0287	.0414	.0534	.0650	.0816
	t	381.6	391.6	401.6	421.6	441.6	461.6	481.6	511.6
200	c	.720	.634	.604	.571	.551	.539	.529	.518
	h	(1198.3)	6.7	12.9	24.7	35.9	46.8	57.4	73.1
	n	(1.5462)	.0079	.0152	.0286	.0412	.0532	.0646	.0811
	t	400.9	410.9	420.9	440.9	460.9	480.9	500.9	530.9
250	c	.740	.649	.617	.580	.560	.545	.535	.523
	h	(1204.2)	6.9	13.2	25.1	36.5	47.6	58.4	74.3
	n	(1.5309)	.0079	.0152	.0286	.0411	.0530	.0644	.0806
	t	417.4	427.4	437.4	457.4	477.4	497.4	517.4	547.4
300	c	.757	.661	.628	.588	.566	.551	.541	.529
	h	(1206.9)	7.0	13.5	25.6	37.1	48.3	59.2	75.3
	n	(1.5186)	.0080	.0152	.0286	.0410	.0528	.0641	.0803
	t	432.0	442.0	452.0	472.0	492.0	512.0	532.0	562.0
350	c	.773	.672	.638	.595	.571	.556	.545	.533
	h	(1213.7)	7.1	13.7	26.0	37.6	48.9	59.9	76.1
	n	(1.5085)	.0080	.0152	.0285	.0409	.0526	.0639	.0799
	t	444.9	454.9	464.9	484.9	504.9	524.9	544.9	574.9
400	c	.788	.682	.646	.601	.576	.560	.548	.536
	h	(1217.7)	7.3	13.9	26.3	38.1	49.4	60.5	76.7
	n	(1.5001)	.0080	.0152	.0285	.0408	.0525	.0636	.0795
	t	467.4	477.4	487.4	507.4	527.4	547.4	567.4	597.4
500	c	.815	.701	.658	.609	.582	.565	.552	.539
	h	(1224.5)	7.5	14.3	26.9	38.8	50.3	61.5	77.8
	n	(1.4863)	.0081	.0153	.0285	.0406	.0521	.0631	.0788
	t	486.9	496.9	506.9	526.9	546.9	566.9	586.9	616.9
600	c	.840	.718	.668	.614	.586	.568	.555	.542
	h	(1230.5)	7.7	14.6	27.4	39.4	50.9	62.2	78.6
	n	(1.4757)	.0081	.0153	.0284	.0404	.0518	.0626	.0781

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM—Continued

DEGREES OF SUPERHEAT										
160	200	250	300	350	400	450	500	600		p
471.8	511.8	561.8	611.8	661.8	711.8	761.8	811.8	911.8	t	
.494	.490	.490	.491	.494	.497	.500	.504	.511	c	80
84.2	103.9	128.4	152.9	177.6	202.4	227.3	252.4	303.1	h	
.0995	.1202	.1447	.1682	.1907	.2123	.2331	.2532	.2916	n	
487.6	527.6	577.6	627.6	677.6	727.6	777.6	827.6	927.6	t	
.498	.493	.493	.495	.497	.500	.503	.507	.515	c	100
85.2	105.1	129.7	154.4	179.2	204.1	229.2	254.4	305.5	h	
.0989	.1193	.1437	.1669	.1892	.2107	.2313	.2513	.2895	n	
507.1	547.1	597.1	647.1	697.1	747.1	797.1	847.1	947.1	t	
.502	.497	.496	.498	.500	.504	.507	.511	.519	c	130
86.4	106.4	131.2	156.1	181.0	206.1	231.4	256.8	308.3	h	
.0980	.1183	.1423	.1653	.1873	.2086	.2291	.2489	.2868	n	
523.2	563.2	613.2	663.2	713.2	763.2	813.2	863.2	963.2	t	
.506	.501	.499	.501	.504	.507	.511	.514	.522	c	160
87.4	107.5	132.5	157.5	182.6	207.9	233.3	258.9	310.7	h	
.0973	.1174	.1412	.1640	.1859	.2070	.2273	.2470	.2848	n	
541.6	581.6	631.6	681.6	731.6	781.6	831.6	881.6	981.6	t	
.511	.505	.504	.505	.508	.511	.514	.518	.526	c	200
88.6	108.9	134.1	159.3	184.6	201.1	235.7	261.5	313.7	h	
.0967	.1166	.1402	.1628	.1845	.2054	.2256	.2452	.2828	n	
560.9	600.9	650.9	700.9	750.9	800.9	850.9	900.9	1000.9	t	
.517	.510	.508	.510	.512	.515	.519	.523	.531	c	250
89.9	110.4	135.9	161.3	186.7	212.5	238.4	264.4	317.1	h	
.0961	.1159	.1393	.1617	.1832	.2040	.2241	.2436	.2809	n	
577.4	617.4	667.4	717.4	767.4	817.4	867.4	917.4	1017.4	t	
.521	.514	.512	.513	.515	.519	.522	.526	.534	c	300
91.0	111.7	137.4	163.0	188.7	214.6	240.6	266.8	319.8	h	
.0956	.1152	.1385	.1607	.1821	.2028	.2228	.2421	.2793	n	
592.0	632.0	682.0	732.0	782.0	832.0	882.0	932.0	1032.0	t	
.542	.517	.515	.516	.518	.522	.525	.529	.537	c	350
91.9	112.8	138.6	164.4	190.2	216.2	242.4	268.7	322.0	h	
.0952	.1146	.1377	.1598	.1811	.2016	.2215	.2407	.2777	n	
604.9	644.9	694.9	744.9	794.9	844.9	894.9	944.9	1044.9	t	
.527	.520	.518	.519	.521	.525	.528	.532	.540	c	400
92.7	113.6	139.6	165.5	191.5	217.6	243.9	270.4	324.0	h	
.0947	.1140	.1370	.1589	.1801	.2005	.2203	.2394	.2763	n	
627.4	667.4	717.4	767.4	817.4	867.4	917.4	967.4	1067.4	t	
.550	.523	.521	.522	.525	.529	.532	.537	.545	c	500
93.9	115.0	141.1	167.2	193.4	219.7	246.3	273.0	327.1	h	
.0938	.1128	.1355	.1572	.1782	.1984	.2180	.2371	.2737	n	
646.9	686.9	736.9	786.9	836.9	886.9	936.9	986.9	1086.9	t	
.534	.527	.525	.526	.528	.532	.536	.540	.549	c	600
94.7	116.0	142.3	168.5	194.9	221.4	248.1	275.0	329.4	h	
.0929	.1117	.1341	.1556	.1764	.1964	.2159	.2348	.2711	n	

marked at the top. The four values grouped together for each condition are as follows:

t = temperature of steam, in degrees fahrenheit.
c = specific heat under constant pressure, generally called c_p, at the particular temperature and pressure.
h = heat required to raise the steam from saturation up to the the temperature t, or through the number of degrees at the top of the column, at constant pressure.
n = entropy of superheating, or the entropy gained with the heat h.

In choosing the intervals between the tabular values, in both directions, the idea was to keep these intervals short enough for the effective use of simple rectilinear interpolation. A column for 550 deg. was originally computed, also lines for $p = 450$ and 550 lb.; these were crowded out in bringing the table to page form, but their absence will cause no appreciable error.

29 The above description does not fully apply to the first column, for zero superheat. This is, of course, the state of dry saturation, where h and n, as just defined, are both equal to zero. In the spaces that would thus be left vacant are bracketed: In the h column, the total heat H of one pound of dry saturated steam, above water at 32 deg. fahr.; in the n column, the total entropy of heating and evaporation, corresponding with the heat H.

30 The writer, desiring to dispense with Greek-letter symbols in thermodynamic formulae, and having other uses for the more obvious E, has adopted N as the symbol for entropy. Letting plain N stand for the entropy of dry saturated steam formed in the usual manner, which is given in the zero column of the table as just stated, we have $(N + n)$ as the abscissa of a point on the isopiestic curve for superheat in the entropy-temperature system of representation.

31 Fig. 7 embodies the result of an attempt to develop a general law from experimental data which are not in very close agreement. The quantity sought does not lend itself to easy and precise determination; and although the experiments here discussed are essentially correct in method, and are far superior in reliability to any which have preceded them, they nevertheless leave an area of uncertainty within which probabilities must be balanced and judgment exercised. Based upon such data, the law may be qualitatively correct, and yet be liable to a certain degree of quantitative error. But even though the absolute values may not be quite in conformity with the undis-

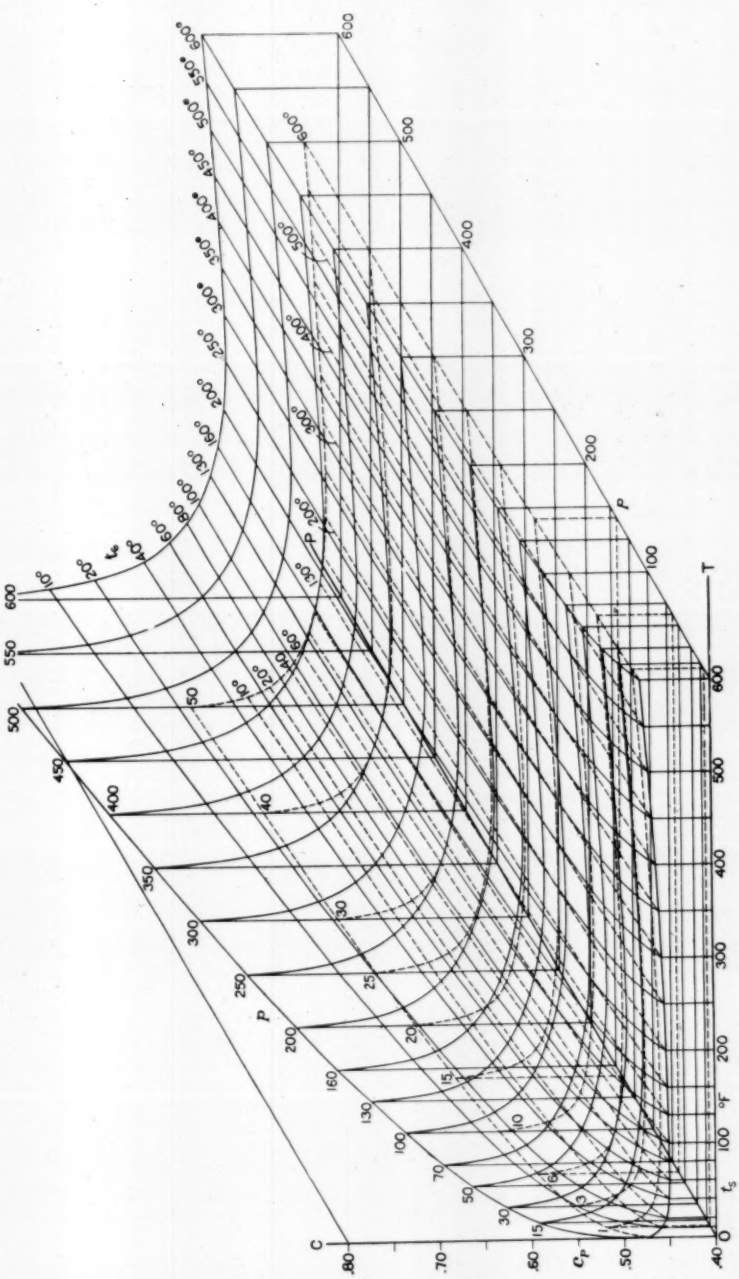

FIG. 7 PROBABLE TRUE SPECIFIC HEAT

covered truth, it is highly important that in their manner of variation they follow some smoothly acting law.

32 The general principles just enunciated are intended to bear upon the question of the degree of precision desirable in the quantities in Table 1. It appears that there must be some definite law of relation between c, t and p, and there is no reason to expect irregularities in the true law. Whether Fig. 7 embodies this true law or not, there is no doubt that a set of curves defines a quantity much less rigorously than does a mathematical formula. It must be acknowledged therefore that the values of c in Table 1, measured from a diagram and accurate to only three figures, are likely to show small irregularities.

33 The numbers h and n were at first computed directly from c as measured, being carried to five-figure accuracy; that is, the heat values had two decimal places, the entropy values five. Checking these up by means of their differences, minor irregularities were found: but by the expedient of making the differences conform to a scheme of regular variation, the values of h and n were brought to a much truer and more exact manner of variation than that of c. Of course, h and n are the actually important quantities for thermodynamic calculations.

34 Although heat and entropy were both worked out to one more decimal place than is given in the table, and together adjusted until the probable irregularity did not exceed two points in this last place, it did not appear that any practically useful purpose would be served by printing the table with more figures than are actually given. In computing the entropy h, values were at first got by plain division, using intervals of 10 to 50 deg., and dividing heat added by average absolute temperature; this first table was then corrected by using more exact methods along a few equally spaced pressure lines, and carrying the differences thus found through the ranges between these closer determinations.

VARIOUS DATA

35 In the preceding discussion, the experiments of Knoblauch and Jakob and of Thomas are accepted as reliable, but no attention is paid to the other experiments, over similar ranges, that have been reported from time to time. Both the "accepted" sets were made by the method of superheating the steam with heat derived from a measured electric current. Knob'auch and Jakob took steam at a temperature t_1 (which ranged from 15 deg. to 370 deg. fahr. above

saturation) and raised it to t_2 (which was from 50 to 140 deg., higher than t_1). Thomas found in every case the heat needed to raise the steam from dry saturation to the particular temperature t. Both took great precautions against radiation, and used effective methods in measuring the unavoidable loss by radiation.

36 The converse of the method of heating is that of cooling, in which heat is abstracted from a current of highly superheated steam and measured by an absorption calorimeter—the steam being still superheated when it leaves the calorimeter. The most prominent experiments along this line were made by Lorenz, and published in the Zeitschrift des Vereins deutscher Ingenieure, 1904, p. 698. The surfaces were water-cooled, hence at a much lower temperature than the steam which flowed over them. The results as to c_p now appear to have been uniformly too large, because of a fact which has come to be recognized only in the last few years. This is, that a current of superheated steam may be far from homogeneous, especially when near to saturation. In the method of Lorenz the outer layers of the steam current may be cooled to saturation and even partly condensed, while the body of the steam is yet superheated and determines the reading of the thermometer. The result is that the amount of heat abstracted is too great in relation to the apparent temperature drop.

37 The other method which has been extensively used is that of throttling or wire drawing. The apparatus is essentially a throttling calorimeter on a large scale, with especial precautions against radiation and conduction of heat. The best experiments have been made by Grindley, Philosophical Transactions of the Royal Society of London, 1900, vol. 194, and by Griessmann, Zeitschrift des Vereins deutscher Ingenieure, 1903, p. 1850. Two main objections lie against this method, one is the difficulty of insuring that truly dry steam is received at the upper pressure; the other, the uncertainty as to whether the throttled and superheated steam is truly homogeneous. In regard to the latter point, Griessmann, who used a porous plug instead of an orifice, would appear to be less liable to error. The writer analyzed Griessmann's results several years ago; and while the individual values of c_p for the same pressure and nearly the same range of temperature do not agree well, the averages of a number of similar determinations check up quite fairly with values from Table 1, as appears in Table 2. Referring to Fig. 6, we see that Griessmann's results would agree better with the curves of Thomas, which are uniformly higher than those decided upon by the writer, at these low degrees of superheat.

TABLE 2 COMPARISON OF GRIESSMANN'S RESULTS

Pressure		Degree fahr. of Superheat	Specific heat c_p.	
Kg.	Lb.		Griessmann	Table 1
1	14	70	0.51	0.487
2	28	45	0.53	0.503
3	43	30	0.55	0.530
4	57	20	0.57	0.550
5	71	15	0.597	0.570
6	85	10	0.62	0.592

38 In this connection, it may be of interest to compare values of the superheat h as given by Thomas with values from Table 1. The values marked T in Table 3 are taken from Fig. 17 of Thomas' paper, which is a plot of the heat quantity h, those marked H are from Table 1. Neither in absolute amount nor in percentage are the differences large enough to be of much practical significance.

TABLE 3 COMPARISON OF SUPERHEAT IN BRITISH THERMAL UNITS

Pressure	Degree of Superheat							
	40		100		200		300	
P	T	H	T	H	T	H	T	H
20	21.8	20.9	51.3	49.9	99.1	97.1	146.1	144.2
60	23.4	22.5	54.0	55.0	103.6	105.1	151.6	151.0
300	25.2	25.6	59.0	59.2	112.5	111.7	161.7	163.0

SUMMARY

39 For a purpose such as the determination of the total heat of formation of superheated steam in the work-up of a boiler test, the results of either Knoblauch and Jakob or of Thomas might be used, within and over the range of actual experiment, without introducing an error of any practical significance. For other purposes, such as calculations from point to point in the field of superheat, the disagreement of the two authorities introduces a very considerable element of uncertainty. The writer has endeavored to balance probabilities between the two, to correct certain manifest errors in interpretation, and to make the law for c_p take a consistent form, in accord-

ance with the rational requirements which have gradually become evident as experimental knowledge in regard to this subject has been enlarged.

40 It is reasonable to believe that the change from the properties of wet steam to those of superheated steam, at the saturation limit, is not absolutely abrupt, but that there is left a little disgregation work, or something closely analogous thereto. This accounts for the higher value of c_p near saturation, as well as for its falling off with the temperature. The writer does not believe that the finally adopted initial cross-curve H_2 in Fig. 2 is at all definitely determined, but he does agree on principle with the experiments of Thomas, which show this rise of c_p toward saturation to persist, although decreasing in amount, into the low ranges of pressure and temperature. An error of 10 or even 20 per cent in the value of the initial c_p would have only a minute effect upon the superheat h.

41 The rise of c_p with temperature, after a minimum has been reached, is a well established fact in the behavior of chemically compound gases, like H_2O and CO_2, and even takes place, though at a very slow rate, with the simple gases such as oxygen and nitrogen. Knoblauch and Jakob fully recognize this fact, indeed over emphasize it in letting rather faint experimental indications make their curves rise too soon and too rapidly. Thomas, on the other hand, ignores it, although his experiments hardly extended far enough into the field of superheat for the upward tendency to have come into action. According to the data on Fig. 5, it appears that the writer's curves are made to rise just a little too rapidly. In laying out Fig. 6 and 7, the idea was that these curves of c_p for different pressures, based upon actual temperature and if projected upon one plane as in Fig. 6, ought to converge very slowly as they rise.

42 In conclusion, the writer would again express his strong confidence in the experiments which form the basis of this discussion; and, at the risk of accusation of rashness, would state the belief that, for all technical purposes, the question of the specific heat of superheated steam under constant pressure is about settled.

CLUTCHES

With Special Reference to Automobile Clutches

BY HENRY SOUTHER, HARTFORD, CONN.

Member of the Society

Clutches of one form or another, if one may judge from the literature on the subject, have been used since the earliest history of the mechanic arts, an important usage dating far back being in connection with wire drawing. It would seem, however, that clutches in general had attracted but little attention in the engineering world until recently when they have been called upon to do the delicate work now required of them in connection with cotton mill machinery, printing presses, electric cranes, power plants and, most recently, with automobiles.

POSITIVE CLUTCHES

2 The positive or jaw clutch is necessarily used only where the character of the starting action is immaterial, and if sudden, matters but little. It obviously can be used only where the inertia of the standing or driven parts is relatively small, otherwise materials could not stand the wear and tear.

3 Modifications of the positive clutch are made in the angles of engagement between the jaws. The least positive form is one where the planes of engagement are inclined backward as regards the direction of motion at an angle of 15 deg., more or less. The tendency of such a clutch under load is to disengage. It must be held up to its work by an axial pressure, which may be regulated to perform a nor-

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mal duty, but to slip and disengage when called upon abnormally by some accident or overload.

4 Positive clutches with engaging planes parallel to the axis of rotation must be held up to their work to guard against a natural tendency to jar out, but they present no safety features against an overload.

5 More extreme yet as to positiveness is the so-called undercut engagement of the jaw clutch, the tendency of which is to engage the tighter when loaded; and which can be disengaged only when absolutely free from load and in a condition to be rotated in a reverse direction sufficiently to overcome the under-cut angle.

6 In automobile construction the positive type of clutch is used inside the gear box, so arranged as to be operated only while the main friction clutch connecting the engine with the driven shaft is disengaged. This positive clutch sometimes takes the form in automo-

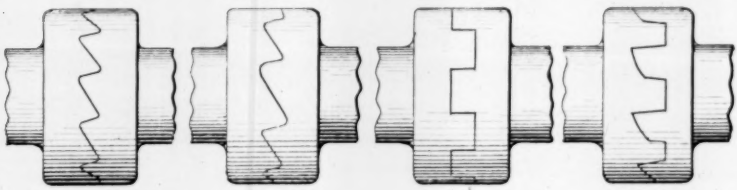

FIG. 1 FIG. 2 FIG. 3 FIG. 4

biles of an external spur gear meshing with an internal spur gear. Automobile gear changing systems are used that keep all gears in mesh all the time. Each gear carries a positive jaw clutch to be engaged with mates on the driving shaft (while the main friction clutch is open).

7 Several inventors of merit have accomplished this same thing by a sliding spline (or hardened ball) on the driven shaft, which engages with the gear hub internally. Such forms are in use, but it can not be said in common use.

8 It will be seen that this use of a positive clutch in connection with the automobile is one where there is little inertia to be overcome, the mass to be started being only a small shaft within the gear box and the gears on it. The chances are that even these are rotating to some extent in the direction in which they are to continue to move. After this positive clutch is once engaged the main friction clutch comes into play. No drag of the friction clutch is permissible.

9 The starting crank of an automobile is a first rate illustration of an under-cut positive clutch. It is under-cut so that when the hand is applied to the starting crank there is little or no danger of the clutch slipping off and wrenching the operator. It is a fact, however, that some of these clutches are not under-cut and are disagreeable to handle for this reason.

CLASSIFICATION OF FRICTION CLUTCHES

10 A rather careful search of the literature reveals the fact that there are basic types, few in number, involved in all clutches, but that there is an infinite variety of detail of construction and manipulation.

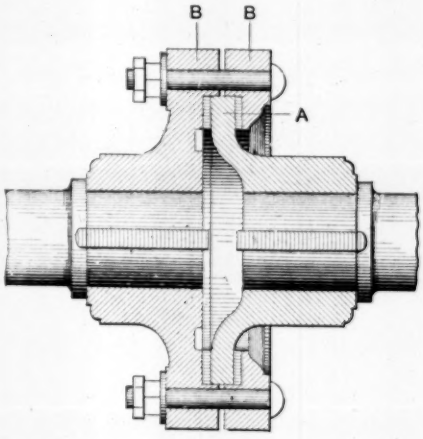

FIG. 5 RAMSBOTTOM CLUTCH

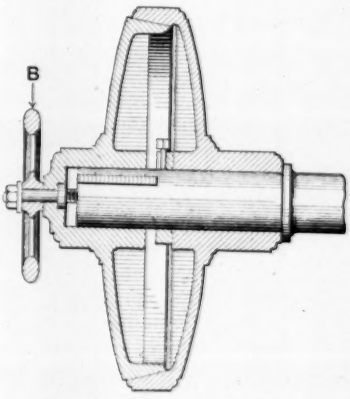

FIG. 6 CONE CLUTCH

11 Rankine differentiates between friction clutches about as follows:

Friction clutch (contracting band),
Friction cones,
Frictional sector (invented by Bodmer),
Friction disc (Weston's invention).

12 Reuleaux illustrates the Ramsbottom clutch as used for rolling mill work. This is nothing more nor less than a friction coupling in which one flange is squeezed between frictional surfaces by being tightly bolted. Referring to Fig. 5, the flange attached to part *A* is firmly clamped beween the wood-lined surfaces of *B*, adjustment of the bolts being such that the friction will resist normal torque but

yield to abnormal torque. This is perhaps the most simple form of friction clutch.

13 It would seem as though some such device might well enter into the transmitting portions of an automobile, so adjusted as to resist up to, say, one-half the elastic limit of the parts involved, and slip under the application of any greater load. Right here it is well to point out, however, that many mechanical devices which have performed well elsewhere have performed badly in automobiles because of the unusually variable conditions to which an automobile is exposed, which might prove to be true of this clutch.

14 Reuleaux then shows as the next step in the development of the clutch a cone coupling, the two parts being forced into engagement by screw and handwheel *B*, as shown in Fig. 6. He states that the angle of the cone should not be less than 10 deg., in order that the

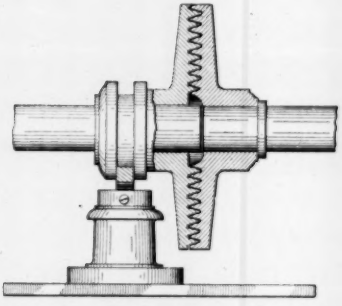

FIG. 7 MULTI-CONE CLUTCH

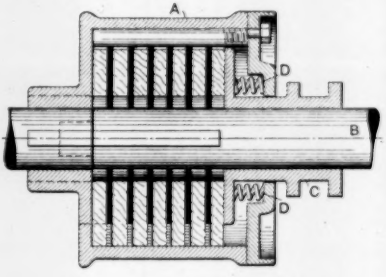

FIG. 8 WESTON CLUTCH

parts may not become wedged together. He also gives in connection with this clutch with frictional surfaces of iron on iron a coefficient of friction of 0.15. In order to keep the axial pressure within reasonable limits, he places the mean radius of the cone between three and six diameters of the shaft.

15 Following the single cone clutch in Reuleaux is what might be called a multi-cone, as shown in section by Fig. 7, a series of concentric cone-shaped rings with angles of 10 deg., or 20 deg. for both halves of the cone. As shown in this cut, it is apparent that the collar would have to resist the pressure and wear due to the axial pressure necessary for proper engagement. This would be serious.

16 Such wear is avoided in heavy machine work or in high-speed automobile design by making the axial pressure self-contained on the rotating member, except when the clutch is in the act of being dis-

engaged. Such construction as this has been found absolutely necessary in connection with all automobile clutches.

17 This modification of the foregoing is shown in Fig. 9. The pressure of the screw wheel is self-contained, the two halves *A* and *B* being clamped together by it, the concentric double-faced cones furnishing much friction at slight axial pressure.

18 The next clutch shown by Reuleaux is that which he attributes to Koechlin, Fig. 10. This is of the internal expanding type, three internal clamp pieces, *A*, fitted with bronze shoes, being thrust out against the enclosing cylindrical drum *B*, by means of lever and screw action. Reuleaux points out the fact that there is no danger of wedging in this clutch, as exists in connection with the cone clutch.[1]

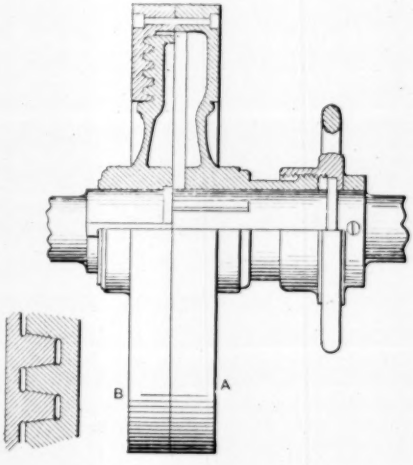

FIG. 9 SELF-CONTAINED THRUST

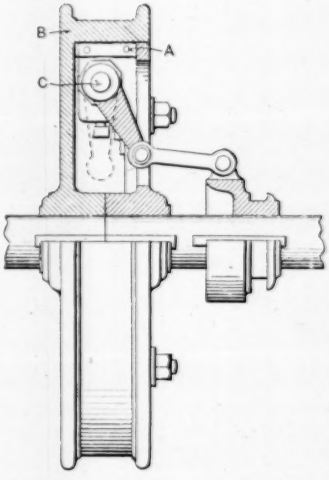

FIG. 10 EXPANDING TYPE

19 Reuleaux next shows a form of "axial friction coupling," well-known as the Weston clutch, based on the principle of multiple plate friction, Fig. 8. The plates are alternately wood and iron, as indicated, the wooden ones engaging with the outside cylindrical containing-case *A*, and the iron ones with the shaft, *B*. In the form shown, the plates are pressed together by springs, *D*, and released by drawing back a collar, *C*, which releases the spring pressure.

[1] Bodmer and Koechlin seem to have been working along similar lines.

MACHINE SHOP CLUTCHES

20 The foregoing references to Reuleaux will serve to fix in mind the fundamental or basic types of clutches and I will now give a number of illustrations to show the development of the machine shop clutch from the earlier forms already illustrated.

21 Perhaps the simplest is the type in which one flat disc presses against another, the surfaces being leather against iron, bronze against iron, or wood against iron, the axial pressure being great enough to drive the maximum load, yet not so great but that slipping takes place when the load is first applied, which prevents all jar. Such clutches are familiar in the driving of looms.

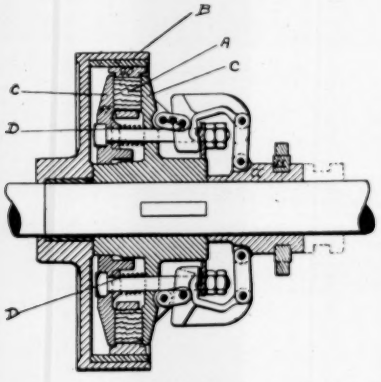

FIG. 11 MODIFICATION OF WESTON TYPE

22 In Fig. 11[1] is a modification of the Weston type. It is not multi-disc, there being only one wooden disc, *A* attached to the enclosing case *B*, which is gripped between two iron surfaces *C*, keyed to the driving shaft. To prevent any drag when disengaged, separating springs *D*, are supplied, which part the frictional surfaces when idle. Slight rubbing when idle is not a very serious matter in machine shop clutches, however, but its importance in connection with automobile clutches, I will bring out later.

23 It is interesting to note that very little information is given or obtainable in regard to the frictional capacity of these machine shop clutches. Correspondence with the manufacturers reveals the fact that knowledge of the capacity of their clutches is of an empirical

[1] Fig. 11 is from the catalogue of T. B. Wood's Sons, Chambersburg, Penna.

character. The coefficient of friction of maple (which is commonly used by them) on cast iron is known. Little use can be made of this knowledge, however, as the degree of lubrication, or lack of it, may easily double or halve this coefficient. The manufacturers have learned by experience what size clutch of their own make is necessary for the transmission of a given horse power. Catalogues usually give the horse power that can be transmitted at 100 r.p.m. It is probable that information of this kind, untechnical though it be, is decidedly more reliable than that obtained from any formula containing an unknown variable—the coefficient of friction.

24 What was formerly known as the Frisby clutch was designed many years ago when no attention was paid to mathematical design, but its capacity has been well established by experience.

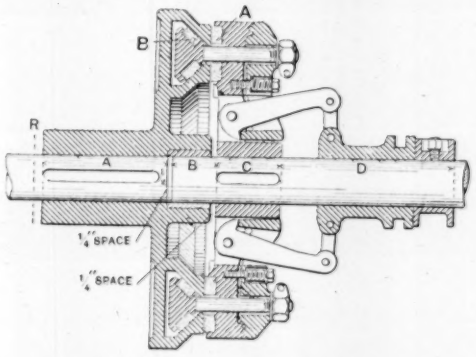

FIG. 12 COMBINED CONE AND FLAT SURFACE

25 Fig. 12 shows this clutch, which is not unlike the last one described, except that a flat surface *A* and cone *B* are used in combination. The gripping of the surfaces is accomplished in very much the same way. The frictional surfaces are separated by springs when disengaged.

26 It is apparent that this clutch would require less axial pressure for any given horse power transmitted than the foregoing type because of the cone; or, in other words, for a given axial pressure would transmit more horse power and therefore would be smaller and more compact, all other things being equal. But here again is the uncertainty of the coefficient of friction. This clutch, for example, might throw its oil to the frictional surfaces more than the previous example, which oil would more than offset the effect of the cone engagement.

27 A firm of clutch manufacturers in the West (Dodge Manufacturing Company, Mishawaka, Ind.) have some experimental data upon the capacity of their clutches which are here given. The results were obtained from clutches fitted with maple blocks and calculations are based on a coefficient of friction of 0.37 and a speed of rotation of 100 r.p.m.

Horse power	Block area	Diameter at block, inches	Circumferential pull at block center	Total pressure	Total pressure per square inch
25	120	16	1960	5300	44
32	141	18	2240	6000	44½
50	208	21½	2900	7800	37½
98	280	27½	4500	12 200	43½

28 A modern adaptation of the old Koechlin form of internal expanding clutch is shown in the following cuts, Figs. 13 and 14 (from catalogue of the A. & F. Brown Company, Elizabethport, N. J.).

FIG. 13

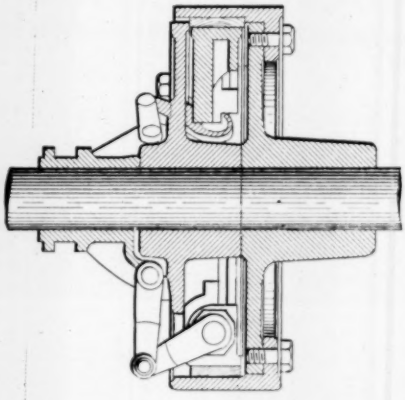

FIG. 14

MODERN ADAPTATION OF KOECHLIN CLUTCH

29 These are largely used for very heavy work, the firm advertising clutches 48 in. in diameter capable of transmitting 320 h.p. at 100 r.p.m. The frictional surfaces are wood, especially prepared for the purpose, against iron; the experts of the company claiming that their experience has shown this combination to be the best. They state that wood against iron is not liable to strike fire, as in cases where both friction surfaces are of iron.

CLUTCHES USED IN WIRE DRAWING

30 One of the oldest usages made of clutches, as already stated, is in the wire drawing art. The iron drum around which the wire is wrapped as a rule contains some form of clutch within it. What I believe to be the most recent development in this direction is from a prominent Eastern engineering firm (Morgan Construction Company, Worcester, Mass.).

31 The clutch developed by them is a compound one, the main driving effort being furnished by a wrapping coil on a chilled iron surface, the initial engagement of the coil being brought about by a modified cone or ring slipping down onto a cone which drags the free end of the coil into engagement. Once seized, the wrapping continues until tight.

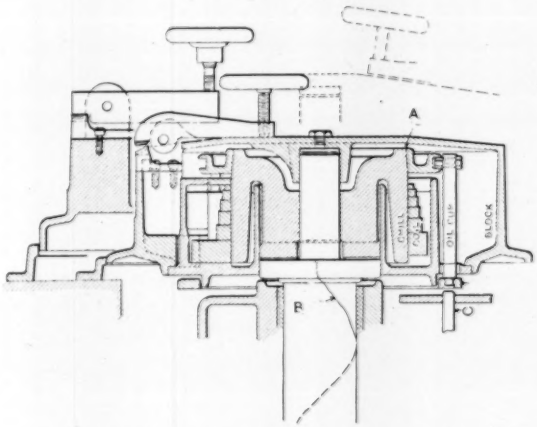

FIG. 15 CLUTCH FOR WIRE DRAWING

32 In Fig. 15 *A* is the tapered friction surface of the chilled drum on which the friction ring bears and below is the coil which is submerged in oil in an annular oil chamber. The drum is 12 in. diameter by 7 in. high and the coil, which is of soft steel, is $1\frac{1}{2}$ in. square at the large, or driving end and $\frac{3}{8}$ in. by $\frac{5}{8}$ in. at the small end. The outside diameter of the block is 25 in.

33 The importance of starting wire into a die gradually and smoothly is very great. I quote from a communication received from the manufacturers of this clutch, which so fully describes the qualities and action of it as to make it a valuable contribution:

Electrically operated wire drawing machines in England have demonstrated that if wire can once be put into motion and the speed increased gradually to prevent breaking the wire, the possible speed of the drawing is almost unlimited. The wire upon which all the work is done becomes exceedingly hot, but the dies remain quite cool.

In order to test the power of our block, we keyed the hub onto the shaft. This hub was flattened on one side. On this flat surface we strapped an 8 in. I-beam about 14 ft. long, and at a distance of 12 ft. 6 in. from the center of the driving shaft we strapped a gear onto this I-beam in which we could put small weights. We first wrapped a $\frac{1}{2}$ in. annealed crane chain around the block, fastening one end to the block and one to the machine, with a weight of 500 lb. at a distance of 12 ft. 6 in. from the center of the driving shaft. The chain broke with a clear fracture. All the links of this chain were strained beyond the elastic limit.

We next took a $\frac{3}{4}$ in. chain and fastened it in the same manner and added weight up to 600 lb., including the weight of the beam, at a radius of 12 ft. 6 in. from the center of the shaft. At this point the cast iron quill which had a bevel gear connected at one end and the chilled friction drum at the other end ruptured, the crack extending from the top of the flange down the spindle a distance of 16 in. in a spiral of $1\frac{1}{2}$ revolutions, as shown at *B*, Fig. 15, the fracture showing clean, close-grained iron. This cast iron quill was 5 in. diameter and cast around a rough turned shaft $2\frac{3}{16}$ in. diameter. It had a flange at the top $8\frac{1}{4}$ in. diameter by $1\frac{1}{4}$ in. thick.

This was the extent to which we carried out experiments and under the above conditions the friction clutch did not slip after it had taken hold.

The following calculation gives the pounds pull exerted on the chain:

$$\frac{600 \times 150 \times 49}{22 \times 12\frac{1}{2}} = 16\ 036 \text{ lb. (more or less)}$$

The pull exerted on the large end of the coil would be equal to

$$\frac{16\ 036 \times 12\frac{1}{2}}{6} = 32\ 761 \text{ lb. (more or less)}$$

The horse power of the clutch at 100 r. p. m. under the above conditions would be

$$\frac{32\ 761 \times 3.1416 \times 100}{33\ 000} = 302 \text{ h.p.}$$

34 A clutch of this kind has been in service some two years, drawing spring wire largely. No repairs or adjustments have been made during that time. The one commercial objection to it is its considerable cost, but it is expected that its good behavior will more than offset that.

CLUTCH OF SMALL DIMENSIONS

35 A strong demand has developed for a clutch of very small dimensions for a given capacity. This demand has been met in

rather a curious way. Instead of cast iron or metal of ordinary strength, hardened tool steel frictional parts have been resorted to. This permits exceedingly high normal pressures between the frictional surfaces. Fig. 16 gives a very good idea of this form of clutch. It will be seen by inspection that the operating collar *A* forces wedge *B* between the long arms of the two levers *C* spreading them in such a manner as to expand a hardened steel ring *D* against the hardened steel enclosing drum *E*.

36 As much as 100 h.p. has been transmitted at 1000 r.p.m. with a clutch containing friction rings $5\frac{1}{4}$ in. in diameter and $1\frac{1}{2}$ in. wide. This form of clutch has been largely introduced into automatic

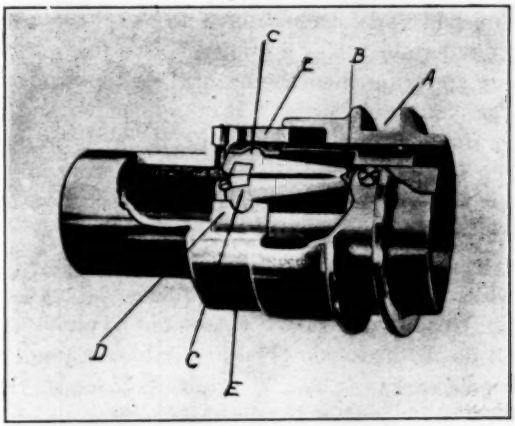

FIG. 16 EXAMPLE OF CLUTCH OF SMALL DIAMETER

machines, machine shop countershafts and launch engines. Its engagement is apparently soft enough for any of these purposes, but in connection with automobile service it is yet in the experimental stage.

CLUTCHES WITH CORK FRICTION SURFACES

37 In connection with the commercial clutches of the forms now under discussion, cork has recently entered the field to a considerable extent and apparently with considerable success. It has a high coefficient of friction, probably double that of wood or leather on iron. Its behavior is peculiar because of its elasticity under compression.

38 As a rule the corks are forced into suitable cavities formed for them in one of the metallic frictional surfaces. The corks are previously boiled and thereby softened and then pressed into the cavities. So established in a metal surface they normally protrude above the surrounding surface and engage first when the surfaces are brought together. If sufficient pressure is applied to the clutch they are forced down flush with the metal surface and act with it in carrying the load. Following the release of the load they again protrude beyond the surrounding metal surface.

39 Two forms of cork are used, one being the cork in its natural condition, the other prepared as follows: Small pieces are compressed into sheets and blocks of any desired shape under very great pressure and under enough heat to cause the natural gums of the cork to exude and act as a binder. This form of prepared cork is really more enduring than the natural, being stronger, firmer and yet possessing much elasticity. It is expensive and has not had wide use for this reason. Nevertheless, it has been most successful in performing service beyond the capacity of other materials. That is, a clutch with cork friction surfaces will carry a greater load than a clutch of the same size of ordinary materials.

40 One example of this I will give, as follows: A Dodge friction clutch carrying 500 h.p. gave much trouble on account of being overloaded. This clutch was strained up as tight as possible and it was all a man could do to throw it. The maple blocks used were replaced with compressed cork. It was then possible to loosen the adjustment of the clutch to such an extent that the operator could throw it with little effort. Following this change it was found that a set of cork blocks outlasted the maple ones five to one.

41 Prof. I. N. Hollis of Harvard University has determined the coefficient of friction of cork on metal. He found that the coefficient of friction for plain cast iron on cast iron is about 0.16; that is, where W represents the pressure on the surfaces and R the frictional resistance,

$$R = 0.16\ W$$

42 Similarly, for plain bronze on cast iron the coefficient of friction is 0.14, or

$$R = 0.14\ W$$

43 The coefficient of friction of the cork on the cast iron, however, was found to be from 0.33 to 0.37, the former, 0.33, being the value for the heavier loads.

44 It is apparent that the coefficient of cork on iron or steel is about double that of iron on iron. It is further claimed that the coefficient of friction for cork is not very much less when lubricated. Cork has much advantage in a moist atmosphere, being very slightly affected by moisture, as compared with maple blocks ordinarily used.

45 Other tests have been made by Prof. C. M. Allen of the Worcester Polytechnic Institute in connection with loom clutches. His results show for a given dimension of clutch a torque for cork inserts nearly double that of a leather faced clutch.

COMPARATIVE TESTS OF LOOM CLUTCHES

TORQUE MEASURED IN POUNDS FEET

Position of clutch	Pressure on clutch pounds	TORQUE	
		No. 1 "compo" clutch with cork inserts	No. 2 leather-faced clutch
Average results of eight positions	89.5	19.50	16.95
Average results of eight positions	151.5	34.20	17.66
Average results of eight positions	213.0	46.43	23.09
Average results of eight positions	275.0	57.05	29.46
Average results of eight positions	337.0	73.33	36.09
Average results of eight positions	398.0	82.24	41.31
Average results of eight positions	460.0	96.48	47.56

AUTOMOBILE CLUTCHES

46 Soon after 1895 the evolution of the automobile or motor vehicle commenced in earnest. There was no difficulty in the way of operating the vehicle with steam or electricity. Positive connection between motor and wheels was quite possible because of the flexibility of the motor.

47 It was realized, however, that these were not the most desirable sources of power. The gas engine in its stationary forms was available. Starting as it does with an explosive impulse, direct connection with the wheels of a vehicle was entirely out of the question. Consequently, a motor vehicle with a gas engine for prime mover was impossible without some means by which the motor and wheels could be separated during the starting of the motor.

48 In May 1879, Geo. B. Selden applied for a patent on a road engine in the United States Patent Office. His application incorpor-

ated the use of a clutch interposed between the engine and the gearing, so as to admit of running the engine while the carriage remained stationary. This is certainly one of the early recognitions of clutch importance in automobile construction.

49 There is little doubt that the appearance of the motor vehicle as a commercial proposition was much delayed by the realization of those skilled in the mechanical art that a good flexible clutch would be difficult to obtain for this purpose. Even at the present time the clutch is as much under discussion among experts in automobile construction as any other element of the automobile.

50 I have searched modern literature on automobile clutches at home and abroad. There are almost as many ideas on automobile clutch construction as there are engineers. This indicates what I have already stated—that the clutch element of the automobile is not by any means a settled one.

51 Violent adherents are found of the cone type, the expanding type and the multiple disc type, and it is interesting to note that all of these types are very old in the art, at least in principle.

52 It would be an endless task to incorporate any considerable percentage of the total information available in a paper presentable before an engineering society. A book devoted to clutch detail alone would be quite possible. The most that I will attempt will be to give a general idea of existing forms of automobile clutches.

53 Theory does not enter into automobile clutch construction to any great extent. References that I have found are contradictory. The theory has been worked out for automobile clutches. I found yards of figures and formulae in some of the German technical papers, also in some of the French automobile journals. The question in my mind is whether this is of any practical use in view of the uncertainty of the coefficient of friction involved. Empirical knowledge seems to be all that is necessary.

54 Perhaps the simplest form of clutch that will be found is that commonly used for small machines and in connection with the planetary system of gear change. This is the pressing of one disc against another, the frictional surfaces being leather, bronze or copper against iron or steel.

55 This form of clutch, Fig. 17, is a very good one as far as it goes, the engagement being soft and gradual. Nevertheless, it is open to very serious objections. If it is so adjusted as to be soft and to pick up its load gradually, a small amount of oil coming between the surfaces renders it absolutely useless. Such a clutch when reasonably dry

will drive a car up a grade sufficient to stall the engine. The same clutch with an over-dose of oil will not drive the car up a half per cent grade on asphalt. This is a good illustration of the uncertainty of figures in connection with clutch design, as far as frictional capacity is concerned.

56 Another form of this same kind of clutch in use in a most successful single cylinder automobile is shown by Fig. 18. I have been able to get the dimensions of this and an idea of the axial pressure necessary, viz:

Maximum radius of leather frictional surface		$4\frac{1}{8}$ in.
Minimum radius of leather frictional surface		$3\frac{13}{16}$ in.
Mean radius of leather frictional surface		$4\frac{5}{16}$ in.
Area of leather frictional surface		$36\frac{1}{2}$ sq. in.
Axial pressure, from		1000 to 1200 lb.
Capacity	Horse power at 600 r.p.m	$5\frac{1}{2}$ h.p.
	Horse power at 1400 r.p.m	10 h.p.

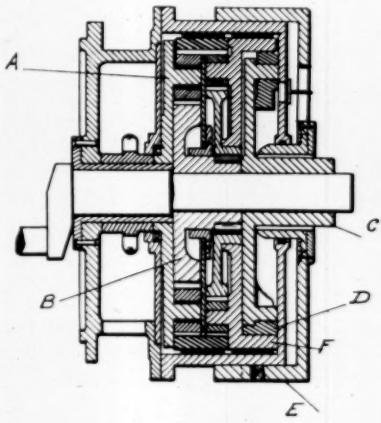

Fig. 17

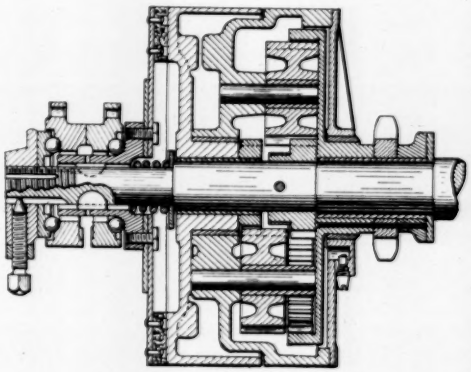

Fig. 18

Types of Planetary Clutches

57 The axial pressure in connection with such clutches is usually furnished by a spring disc; that is, the steel plate which carries the frictional surface, either leather or copper, is caused to operate like a diaphragm spring. The amount of normal surface pressure is not known in most cases as far as I have been able to learn. The above figures in this regard were furnished by the Cadillac Motor Car Company, of Detroit. The diameter of a clutch of this kind ordinarily used to propel a car of 7 h.p. or 8 h.p. is from 5 in. to 10 in., the rub-

bing surfaces being about from one-half to three-quarters of the entire superficial area of the disc.

58 Such clutches are mostly available for two-speed cars, the disc clutch connecting with the engine direct and running at engine speed, the planetary system being used only for low speed and reverse work, actuated by contracting band clutches.

59 Motor vehicles so geared have their uses, but early in the development it was found that three or four speeds were desirable. Boxes of sliding or change gears were resorted to, and here the character of the clutch became of prime importance. To be satisfactory, an automobile clutch used in this manner must engage and disengage easily, requiring but small axial movement of the operating mechanism, or of the clutch itself. It must be entirely independent of centrifugal force, and able to slip for a reasonable length of time without being destroyed.

60 The matter of absolute disengagement is perhaps the most important. Without it the sliding gears intended to be operated when the clutch is free or disengaged cannot be unmeshed nor remeshed. The slightest drag or friction in the clutch means a savage clashing of gears when changed. This means the destruction of the gears and the failure of the entire system of gear change.

61 The early history of the art is full of failures in the matter of successful construction and operation of the so-called sliding-gear transmission. Gears with the teeth worn away were the rule rather than the exception. This wearing was, no doubt, due to the imperfect disengagement of the clutch.

62 The use of a system of gear change requiring the clashing of moving gears cannot receive the complete approval of the engineering world; yet this system has become a success by a combination of improvements in clutch and materials of which the gears are made, and treatment of the materials.

63 An important feature in the clutch is the question of its weight, especially as affecting its inertia. A clutch having high flywheel effect spins to such an extent as to cause violent clashing of idle gears. Consequently, clutches are made as light as possible, and the smaller in diameter the better. Aluminum enters largely into clutch construction for this reason.

64 The spinning of the clutch has been met in many automobiles by a so-called clutch brake—a retarding finger which operates in connection with the clutch disengaging lever and bears upon some portion of the driven member of the clutch, braking it to a standstill.

65 One inventor has gone still further in this direction and disengages the driven shaft both before and behind the gear box. This invention is such that the friction clutch opens first, immediately followed by the opening of a positive jaw clutch behind the gears and then the braking of this disconnected driving shaft, as just described. This permits the engagement of gears that are absolutely free and stationary. It may be well to bring out the fact, however, that this invention is open to one objection, and that is that the gears may stop in such a position as to make it difficult to mesh them.

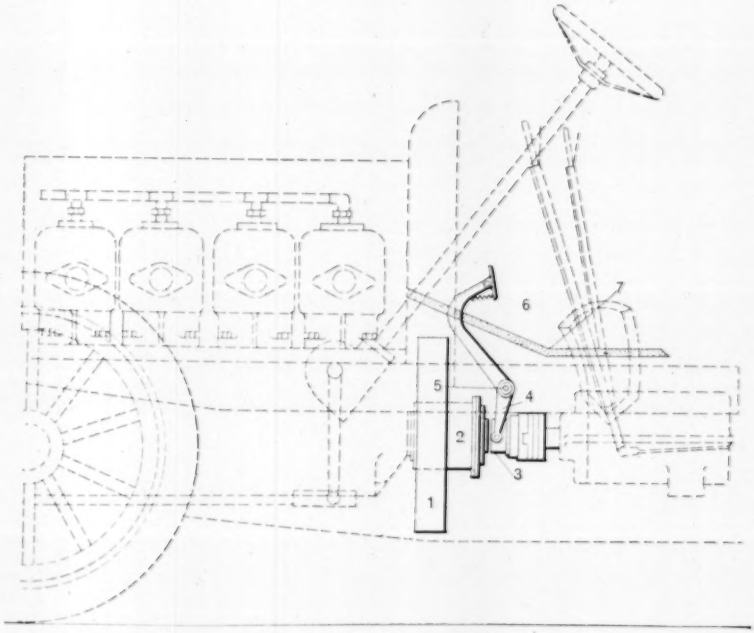

FIG. 19 LOCATION OF AUTOMOBILE CLUTCH

66 Any automobile clutch must engage smoothly and absolutely without shock to be called a success. The quicker it seizes without shock the better it is. Clutches exist that can be engaged suddenly and still not jar the passengers. But such a clutch is open to one very serious objection; that of not picking up the load quickly enough on a hill to start the car forward after a change of gears, before the momentum of the car is materially lessened. For example, in changing from the high gear to the next lower on a steep hill, a clutch that is too soft will permit the speed and momentum of the car to drop

to such an extent that when the clutch finally does take hold the car is nearly at a standstill. This necessitates a further drop into a lower gear; one that will start a car from a standstill on a hill. The clutch designer is, therefore, between two fires; too little slip on one hand and too much slip on the other. A degree of slip between the two must be found, and once found be capable of being maintained. It is doubtful if such a problem exists in connection with clutches anywhere else in the mechanical art.

67 The customary location for an automobile clutch is fairly shown by Fig. 19, that is, within the flywheel or at least at the rear end of the engine if the flywheel is at the front end.

68 This figure shows the application of the multiple disc type with very little room between the gear box and the clutch, and only an Oldham coupling to give flexibility. This would be too close construction (as will be shown later) for the application of the cone type of clutch, which requires so much flexibility back of the clutch. This illustration gives, however, a very clear idea of automobile clutch location in general, and the numbered parts are as follows: 1, flywheel; 2, cylinder enclosing clutch; 3, clutch shifting collar; 4, lever operating sliding collar; 5, engine base; 6, foot pedal; 7, floor boards.

THE CONE CLUTCH

69 I will take up the simplest form first, namely, the cone. I am pretty well satisfied, that, all things considered, it is the best form when properly designed and mounted. It has the advantage of engaging and disengaging with very small axial motion. Axial pressure may be low because the normal pressure between frictional surfaces is multiplied by the angularity of the cone. Its weight may be very small, as the male member may be of aluminum. Its engagement is entirely independent of speed and centrifugal force. No liquid lubricant is needed with attending viscosity, drag and change due to wear and temperature. Disengagement may, therefore, be made perfect.

70 Proper engagement, however, has proven to be a very difficult and baffling problem. I think it safe to say that this difficulty has caused nearly all the rejections that have occurred of the cone clutch. A cone clutch may operate almost as savagely as a positive jaw clutch. It may also refuse to engage, if it does not have the proper combination of angularity, pressure and lubrication. It may behave well at times and very badly at other times. A cone clutch of given angles

and dimensions, with a definite axial pressure, may be a success in one car and an absolute failure in another.

71 The cause of this contradictory behavior may not be and often is not in the clutch proper; but, on the contrary, in the surrounding mechanism of the clutch. The cone clutch must be absolutely free to center itself and seat itself uniformly. A short Oldham coupling or a single universal joint between the clutch and the driven shaft of the car is not enough to permit this under all conditions.

72 The weaving of the frame of the car puts cross strains on such a coupling, causing it to bind and causing the clutch to seize on one side before the other and be drawn suddenly into full contact. A change of angle, increased lubrication and a change of materials on the friction surface will not remove the trouble arising from this cause. A pair of generous, free working universal joints must be provided, in order that the cone shall reach its seat as intended.

73 Similarly, an engine mounted on a flexible sub-frame or pan support may move sufficiently to prevent the proper seating of the cone and cause a similar line of troubles. The male member must be mounted so as to be flexible enough to follow such small movements.

74 Experience has been a long time in teaching engineers that so much trouble can arise from apparently so small a cause; yet there are cases where misbehaving clutches have become well-nigh perfect clutches by the introduction of flexible couplings.

75 Leather (riveted onto an aluminum cone) usually forms one of the rubbing surfaces and gray cast iron the other. It is desirable that the leather shall be kept soft by neatsfoot or castor oil. Some builders boil the leather in tallow before applying to the clutch surface; others do not, but this matter is of minor importance compared with the mounting.

76 With leather $\frac{1}{4}$ in. to $\frac{3}{8}$ in. thick, properly softened, engagement may be sufficiently mild, but an improvement is obtained by placing under the leather at six or eight points on the periphery of the cone flat or spiral springs that cause the leather to engage at these points a little bit before engaging elsewhere.

77 In La France Automobile I find reference to an unusual arrangement of springs underneath the leather of a cone clutch. It is shown very plainly by Fig. 20. It is apparent that the metal of the cone is entirely cut away for a short section, admitting a flat spring not more than an inch wide. The unusual feature is the use of a pair of spiral springs supporting the two ends of the more or less flexible flat spring. I have not seen this construction in use but it ought to be a particularly good way of accomplishing this object.

78 In some instances rubber buffers have been used under the leather in place of springs.

79 The other frictional surface bearing against the leather is, as a rule, a cast iron flywheel.

80 It is obvious that the construction surrounding the clutch must be such that by no means can an unusual supply of lubricant find its way to the frictional surfaces of the clutch. The flywheel prevents any oil from the engine working its way back, being provided with oil trap grooves for that purpose if necessary.

81 From the other direction, the gear-box, for example, oil ordinarily does not get as far as the clutch. There is usually a considerable space between the clutch and gear box. In general, it may be stated that the cone clutch is as free from variations due to lubricants as any other. The leather surfaces gradually become dry and hard, requiring the application of castor or neatsfoot oil preferably, but not very often.

FIG. 20 SPRING ARRANGEMENT

82 With proper usage, cone clutches with leather faces seem to last indefinitely. I have accurate knowledge of cars that have surely been driven 20 000 or 30 000 miles without replacement of leather on the cone face. My own experience is confirmatory of this. I have driven several cars with cone clutches and have yet to experience any trouble from the wearing of the leather face. I recently saw a clutch that had been used for about 2500 miles by me that gave no evidence of wear, neither had it received any attention on my part, except to dose it a few times with neatsfoot oil.

83 There is one defect in the operation of the cone clutch that has caused considerable trouble. The clutch necessarily requires some end or axial motion and a slip-joint that will permit it. An ordinary square slip-shaft has been commonly used. Instances have been found where these square slip-shafts have jammed under load and seized, so as to refuse to permit of the disengagement of the clutch at critical moments. This is a very serious objection, and one that has not been altogether satisfactorily overcome. Improved materials, increased dimensions and better facilities for lubrication have cured much of the trouble. Generous feathers and splines have

been resorted to, which present working surfaces that are normal to each other and which avoid any cam-like or wedging action which may exist with a square shaft bearing in a reasonably easy fitting square hole. Here, again, the perfect freedom introduced by double universal joints plays an important part, the square shaft being very much less apt to bind when perfectly free to center itself.

84 There has been a considerable variety of opinion as to the proper cone angle. Various authorities have placed it all the way from 7 deg. to 20 deg. The French have settled down on an 8 deg. to 9 deg. angle as being about right for a leather faced cone. Several important American makers are using 12 deg. to 13 deg., several 10 deg., and others 8 deg.

85 The following table gives the dimensions for cone clutches used on three different models which are probably as successful as any:

Area of flywheel	113.1 sq. in.	78.7 sq. in.	73.59 sq. in.
Angle (one side)	8 deg.	8 deg.	8 deg.
Radius (maximum)	8½ in.	8¼ in.	7⅞ in.
Spring pressure	375 lb.	320 lb.	250 lb.
Horse power by A.L.A.M. Formula	48	42	40

86 The metal-to-metal cone clutch is a good one. It may be made smaller in diameter and with a sharper angle, say, 7 deg., without seizing. It may be used in connection with copious lubrication. This form has been and is used only to a small extent. The dividing line between slipping and seizing is narrow.

87 Another form of cone clutch has an aluminum male member of about 12 deg. angle bearing against cast iron and with cork inserts in the face of the male member. This clutch is not easily affected by a lubricant and, in fact, may be run with copious lubrication. This type has not been widely enough used yet to give sufficient knowledge as to the possibility of general application under many varying conditions.

88 Up to this time I have referred entirely to what may be called a direct-acting cone, one where the male part of the cone moves axially towards the engine. This is well illustrated by Fig. 21, which is about the simplest form of leather-faced cone clutch.

89 Modifications of this are many, Fig. 22 showing a clutch of the same principle, but in place of having one strong actuating spring

surrounding its axis, it has three weaker spiral springs nearer the periphery of the male member.

90 Fig. 23 is a clutch used for a 50 h.p. car, with a cone angle of 13 deg., a diameter of about 16 in., a total frictional area of about 128 sq. in, and axial pressure of 375 lb. resulting from spring. This cut clearly shows a small spiral plunger spring, *A*, underneath the leather face, *B*, to make it pick up its load more quietly and smoothly. This cut also shows a form of slip-joint back of the clutch, *C*, which, although it does fairly good work, is not on the whole as satisfactory as the double-toggle universal joint. It will be noticed that the arms of this joint have been spread as widely as possible, but, at the best, the pressure and binding action is considerable.

91 In direct contrast to this clutch is the one shown in Fig. 24

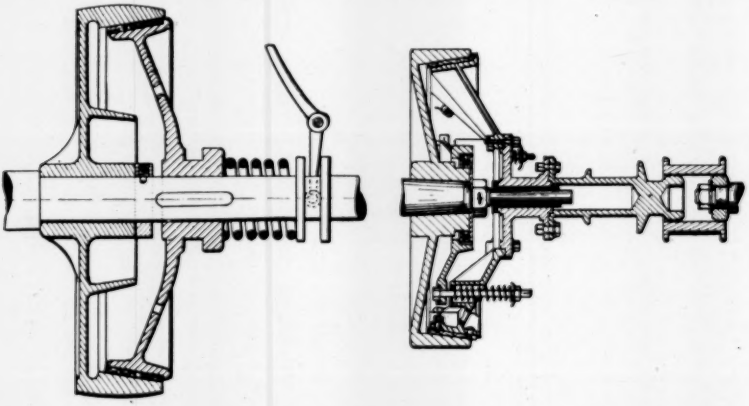

FIG. 21 FIG. 22

LEATHER-FACED CONE CLUTCHES

where the diameter of the cone is very much less, not to exceed 10 in. (the exact dimension I have not, but know it to be about that). This is a clutch used in connection with a car developing 30 h.p., A.L.A.M. rating, and one that has at times developed much higher horse power on the block—as high as 36 h.p. The clutch angle is 13 deg. and the frictional area the first two years this car was built was 86 sq. in., but this has recently been raised to 96 sq. in., the spring pressure remaining at 400 lb. It will be noted at the bottom of this cut that there is a sketch showing the spiral spring plungers underneath the leather.

92 Fig. 25 shows an early form of cone clutch used about 1902 or 1903 for a car of about twenty horse power, This has multi-springs

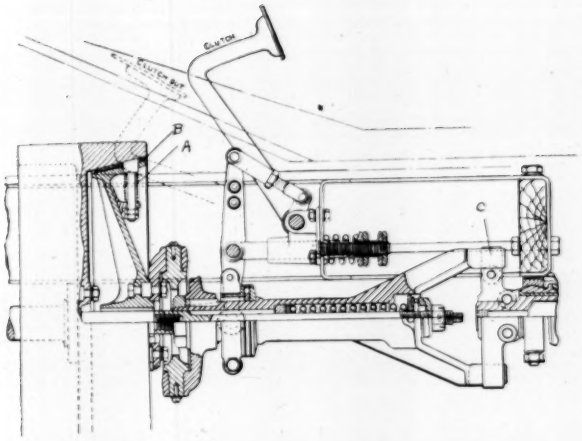

FIG. 23 CONE CLUTCH FOR 50 H.P. AUTOMOBILE

for creating the proper frictional contact and a peculiar form of spring application, simple in the extreme. One of the early forms of toggle joint is also shown at *A*. This gave in its day what was considered very good service.

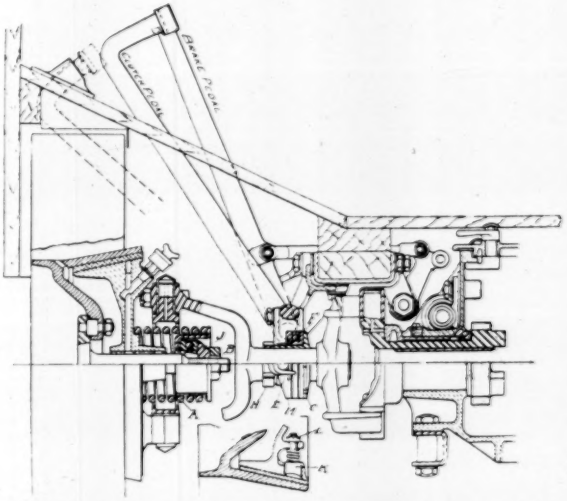

FIG. 24 CLUTCH OF SMALL DIMENSIONS FOR 30 H.P. CAR

MODIFICATIONS OF CONE CLUTCH

93 In the Commercial Motor for October 31, 1907, p. 218, is shown what may be called a multi-cone clutch. This is seen in Fig. 26. The explanation, to be as simple as possible, is that when the clutch engages, the smallest cone seizes first, commences to revolve

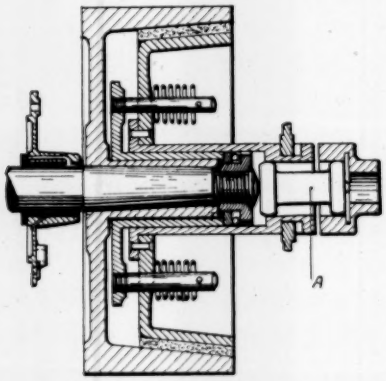

FIG. 25 EARLY FORM OF CONE CLUTCH

and subjects the spiral spring between the next two clutches to torsional movement, which draws them together and brings the two outer cones into action; the idea being that the small clutch shall slip, tend to accelerate the car, that the medium clutch shall behave in a

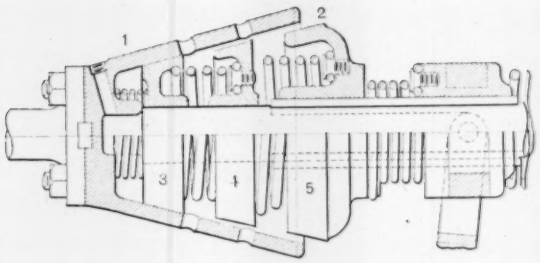

FIG. 26 MULTI-CONE CLUTCH

similar manner and that when the large clutch comes into play the three combined pick up the load and move the car. As far as I know, this has not been tried out sufficiently to say whether or not it is a practical success, but it is interesting in showing the amount of thought that has been given to cone clutches.

94 I have said that theory did not enter into the clutch very much, but below is Fig. 27, which shows the peculiarity of a simple clutch embodying the tractrix curve. This curve is adopted because it is of such a form that by the figured relation of pressures and peripheral speed, wear ought to take place uniformly at all points regardless of the distance of the point from the center. The claim is made for it that the clearance required to complete the engagement is very small; that there is no wedging action between the two members of the clutch and that there is no chance for it to bind. Also that it is simple and particularly adaptable to metal-to-metal clutches.

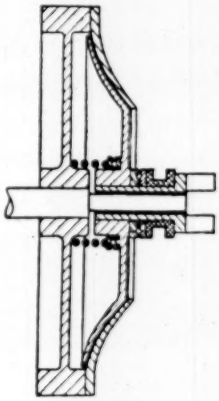

FIG. 27

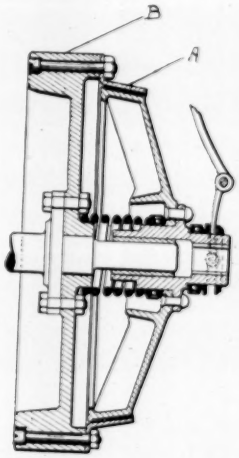

FIG. 28

MODIFICATIONS OF CONE CLUTCH

95 It is in effect a flat disc clutch which will not wear faster near its outside edge than near its inside edge, but beyond that I see no gain. It would certainly require very heavy axial spring pressure, just as a flat disc would. The matter of wear is of little moment, either with flat discs or cones.

96 The so-called inverted cone is well illustrated in Fig. 28. The reversed cone is contained in an extension, *A*, built onto the flywheel *B*. When the cone is disengaged it moves toward the engine, exactly reversing the action of the foregoing type. This clutch has its adherents, and it is a good one, differing very slightly, if properly assembled, in its efficiency from the direct acting cone. It may be kept free from dirt and oil much more perfectly than in the other form.

97 A simple formula for calculating the ordinary cone clutch

is the following, by Chas. H. Schabinger, taken from The Horseless Age of October 2, 1907:

$$h.p. = \frac{P f r R}{63\,000 \sin \theta}$$

P = Assumed pressure of engaging spring in pounds;
f = Coefficient of friction, which in ordinary practice is about 0.25;
r = Mean radius of the cone in inches;
R = Revolutions of the motor per minute;
$\sin \theta$ = Sine of the angle of the clutch.

98 To obtain the size of spring when the horse power is known, the following formula may be used with good results:

$$P = \frac{h.p.\ 63{,}000 \sin \theta}{f\ r\ R}$$

the same symbols being used as in the preceding formula.

99 It will be noted that the coefficient of friction used is 0.25. This is probably near enough for a properly lubricated leather-iron clutch.

EXPANDING BAND CLUTCH

100 The next type of clutch may be classified as internal expanding band or ring. This has had many exponents in the automobile art, but is open to centrifugal effects to such an extent that it requires considerable ingenuity to overcome troubles arising therefrom. At high engine speeds the operating levers have in many cases been so arranged as to lower the normal pressure between the frictional surfaces, resulting in a slippage and arbitrarily fixing a maximum limit of speed for the car on the high gear and of horse power possible to develop in low gear.

101 Fig. 29 shows a clutch operating on the same principle, driving a 16 h.p. car, the spring pull being 50 lb., the diameter of the clutch about 9.50 in., and the width of the band 2 in.

102 This clutch was a particularly soft operating one, but did release at high engine speeds. It operated best with a certain definite quality and quantity of lubricant, which, if varied to any great extent, caused a slipping clutch or a sharply biting clutch. The tendency of the clutch is to unwrap and expand against the enclosing cylinder as soon as any friction is applied to it.

103 The successor of this clutch is shown in Fig. 30, so designed as to overcome the centrifugal releasing effect of levers in the clutch

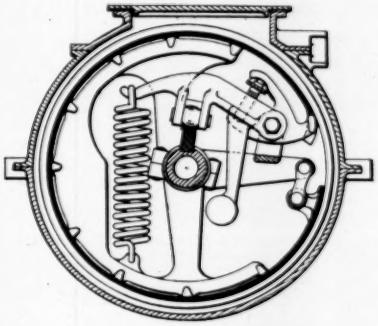

FIG. 29 EXPANDING BAND

shown in Fig. 29. The total area of the clutch is 36 sq. in. and the two springs are of 125 lb. tension each. This clutch was a success, but was finally given up in favor of a simple cone.

FIG. 30

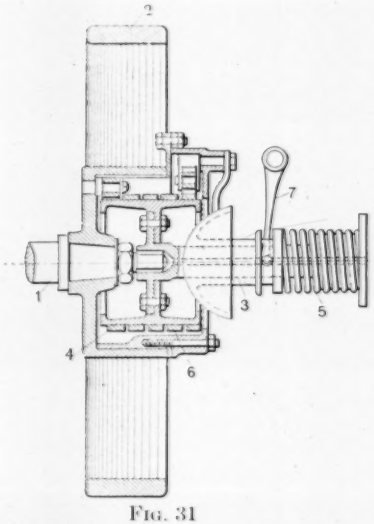

FIG. 31

MODIFICATIONS OF BAND CLUTCHES

CONTRACTING BAND CLUTCHES

104 The exponents of the contracting band type of clutch are few and far between, unless the contracting spiral be so classed, and per-

haps it ought to be. Fig. 32 and 33 show a contracting band characteristic of one of the prominent French cars (Mors) A leather lined flexible steel band (*8*) contracts against a steel cylindrical band (*2*) bolted onto the flywheel (*1*). Clutches of this character are seldom found in the automobile industry, except in two-speed cars.

105 About 1897 a single cylinder, 10 h.p. car was equipped with such a clutch as shown in Fig. 33, a leather lined band, very flexible in character, wrapping around the hub of a flywheel and tightened with a spring pressure of about 50 lb. against a wedge. In this clutch a weight was furnished which would throw out at high speeds and further tend to tighten the band about the hub of the flywheel. The fact that this clutch has not had any successors is an indication that it could not compete with other forms; nevertheless, it was a successful clutch, especially for its time.

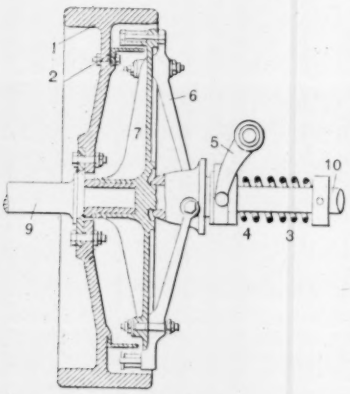

FIG. 32

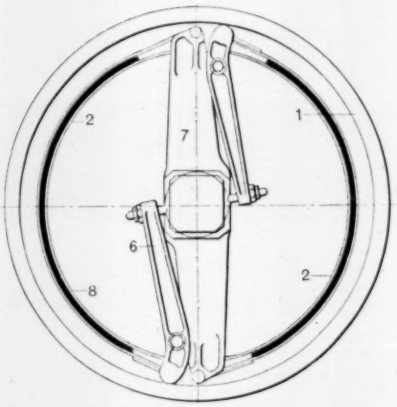

FIG. 33

CONTRACTING BAND CLUTCHES

106 Fig. 31 shows a form of clutch that has had prominent adherents. It is the wrapping spiral spring, of either hard or soft metal. The cut indicates the spring in cross section, marked "*6*," wrapping on the drum *4*. Probably the greatest enemy to this clutch has been the adjustment of the clutching force. With too little lubrication these clutches grip too savagely. With too much lubrication they will not pick up their load rapidly enough. The margin is narrow and hard to control. With a viscous lubricant there is enough drag to make the gears clash badly. The disengagement is not very complete at the best.

DISC CLUTCHES

107 The single disc clutch is widely used, both here and abroad. It is so characteristic of a French make as to travel under the name of the firm—the De Dion. It is now used in this country by one firm for horse powers ranging from 70 to 20, for pleasure and for commercial service. The clutch has a disc *A*, Fig. 35, on the driven member, *B*, which is clamped between two discs, *C*, on the driving member or flywheel. In Fig. 35 this arrangement is clearly shown. There are the necessary accompaniments of separating springs, so as to make disengagement perfect, also either single or multiple springs to cause the proper engagement.

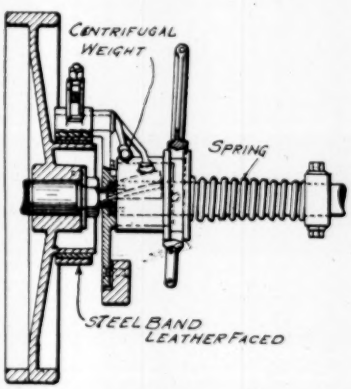

FIG. 34 WRAPPING BAND

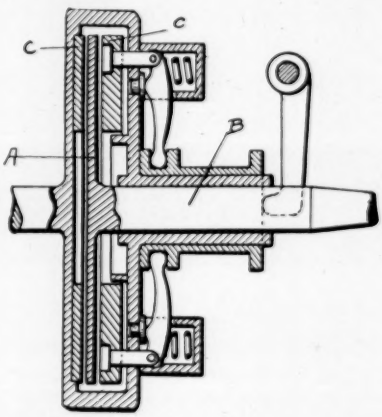

FIG. 35 SINGLE DISC

108 Fig. 36 shows the same kind of a clutch in a slightly different form. The springs in this case are on the front side of the flywheel rather than on the rear, as in Fig. 35. Cork inserts are being used in this clutch to considerable advantage.

109 Another form is the now popular multi-disc clutch; that is, the elaboration of the Weston clutch, to which I have already referred. This clutch is indicated clearly by Fig. 37, the alternate plates of bronze and steel attached to driving and driven parts being pressed together by a powerful spring, *5*.

110 The question of lubrication here is the all-controlling one and, in fact, it would seem that the principal problem in connection with the multi-disc clutch as a type is the proper lubrication of it.

111 I have ridden in cars equipped with such clutches that were extremely savage in taking hold. I have ridden in others of the same make that were extremely slow to take hold. In a way, this may be a good thing. For example, a person going into a hill climbing contest or race and wishing to pick up quickly would be perfectly willing to put up with a harsh clutch and lubricate accordingly. On the other hand, a car running about a level city, encountering few bad hills, would be able to lubricate excessively and still have a satisfactorily driven automobile. A clutch so lubricated would be extremely soft, and yet pick the car up fast enough for ordinary purposes on level roads.

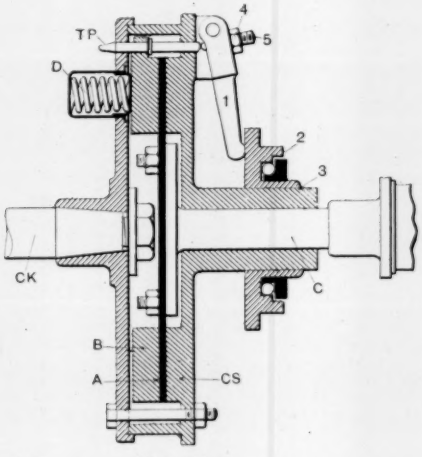

FIG. 36

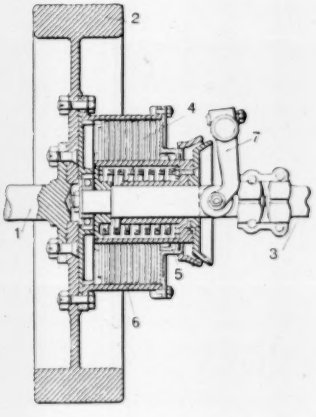

FIG. 37

TYPES OF DISC CLUTCHES

112 Cold and heat affect the operation of this clutch, the lubricant in summer being thicker than can be permitted in winter. As it runs in oil it takes a certain length of time for the oil to squeeze out when engaged and for the metal to come in contact with metal and really begin to drive the car. It will be seen from this that the viscosity of the lubricant is of prime importance.

113 One form of multiple disc clutch in use in a very high grade car consists of steel discs rubbing against a special bronze rolled into sheets. The steel discs are provided with several small tongues on the outer periphery, bent one side sufficiently to come in contact with the next steel disc, for the purpose of separating the discs and over-

coming the drag when the clutch is disengaged. A small clutch brake is also provided to overcome clutch inertia or drag inherent in the clutch and due to viscosity of lubricant. The steel discs are put into the clutch as received from the rolling mill, with the hard black finish characteristic of carefully finished crucible sheet steel.

114 This clutch is connected with the crank case so that oil feeds into it from the crank case through a hole drilled in the center of the crank shaft. Entire reliance, however, is not placed on this supply, a little extra oil being supplied every two or three days through holes provided for that purpose.

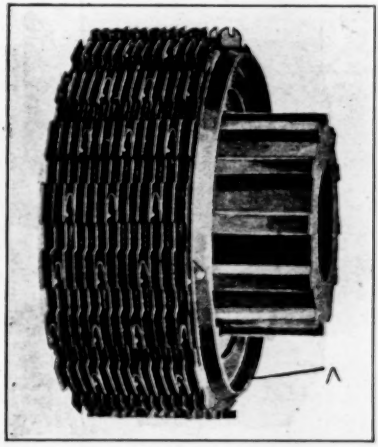

FIG. 38 MULTI-DISC CLUTCH

115 Fig. 38 shows a nest of discs such as used in a well-known multi-disc clutch. The U shaped separating springs are plainly visible. These force like discs apart when the spring pressure is released, overcoming the natural tendency of the oil to cause them to adhere. Some disc clutches are forced apart in a similar manner by little spring shaped strips struck up from the discs themselves.

116 Another form of this same type of clutch is shown in Fig. 39. Comparatively few discs are used as will be noted. On the other hand, it is apparent that the spring pressure is very heavy. This is a successful and well behaving clutch used on a popular car at the present time. It drags but little when the gears are changed and is satisfactory in that respect.

117 A type of disc clutch consisting of all steel discs with alternate ones faced with leather was operated without any oil whatsoever, the leather being softened and made more or less pliable like the leather on the simple cone clutch. These clutches gave some trouble by burning up, the slip required to start smoothly being also sufficient to create enough heat to destroy the leather. This clutch was, however, extremely efficient in the transmission of power. For example, the one shown in Fig. 41, the discs of which are about 7 in. in diameter, was powerful enough to drive an automobile of 50 h.p.

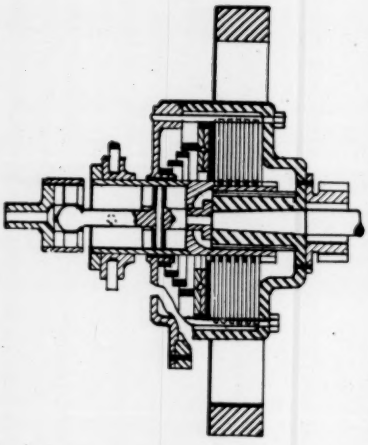

Fig. 39

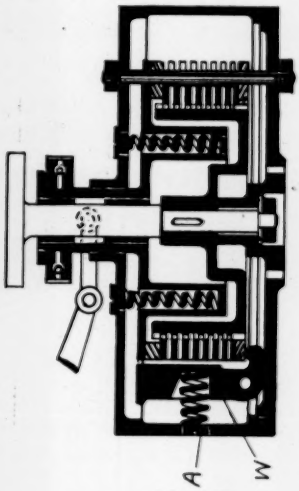

Fig. 40

Types of Multi-disc Clutches

118 It must be remembered that the automobile engine runs at high speed, say, 1000 to 1200 r.p.m. when developing anywhere near its normal rating. Some motors, in fact, running up as high as 1500 to 1800 r.p.m. (standard rating is at 1000 ft. per minute piston speed).

119 It is a fact that in service cars with disc clutches of this character vary more or less in the way their clutches behave. Clutches receive very much less attention than they ought, like everything else on the automobile. I think it will be admitted, even by the adherents of this form of clutch, that it ought to receive more attention than the leather faced cone. Nevertheless, this is now a very successful type of clutch, largely used in many high grade cars.

120 In the matter of the number of plates in the disc clutch there is no agreement between designers. Some use a very large number of thin plates, as many as 50 or 60, and others use a very small number, as few as six or eight; in fact, it may be said that the single disc clutch, which has only two frictional surfaces, is the lower limit.

121 One very ingenious application of the multiple disc clutch has been made by a manufacturing concern in the East (Sturtevant Mill Company, Boston) in the fact that the pressure on the discs is brought about by centrifugal force acting on weights so arranged as to press the tighter with increased velocity. This is shown by Fig.

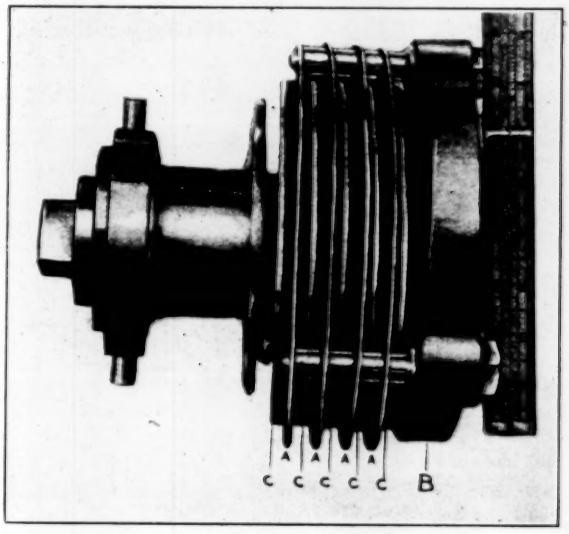

FIG. 41 MULTI-DISC CLUTCH

40. One of the weights is at *W*. It will be noticed that this weight operates against a spring, *A*, which prevents it flying out and gripping at too low an engine speed. Once, however, this spring pressure is overcome, the discs indicated by the alternately light and dark spaces are pressed together.

122 It would seem that this principle has one serious defect in the fact that at low engine speeds the gripping tendency would be small. It would, therefore, not be possible to develop high torque at low speeds, which is sometimes quite desirable. It is a fact, however, that it is almost impossible to stall an engine by applying this

clutch too quickly; it does its own releasing so promptly and automatically. It is almost human in this respect.

123 This principle has been elaborated in connection with an automatic change of gears: gear No. 1 being picked up at a given rate of revolutions by its set of disc clutches; gear No. 2 by an increased number of revolutions by a separate set of discs, and so on. In driving a car so equipped the changes take place without being perceptible except with the closest observation.

124 This system is open to the objection, however, of not being able to spin the engine up very rapidly and connect with the low gear, in order to jump the car out of a hole or some unusual situation. I understand this has been overcome by supplying an independently operated lever for the foot to be used in emergencies only.

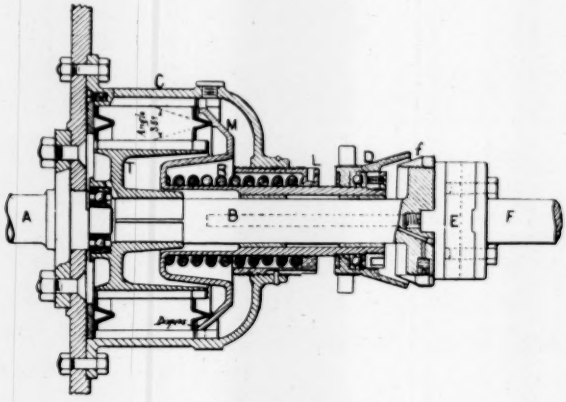

FIG. 42 V-SHAPED DISCS

125 With the automatic Sturtevant multi-disc clutch it has been found experimentally that, for the maximum slip speed usual in automobiles, 15 to 20 lb. per square inch pressure is safe, and that lubricated cast iron discs scarcely wear out the tool marks after many thousands of miles use. They further state that experience has shown them that safe slip is merely a matter of good lubrication and low pressures. They have experimented with small cast iron discs, running dry and with constant slip at two pounds per square inch pressure, and even at that they wore many weeks transmitting a heavy load.

126 A modification of the multi-disc clutch in which the cone and the disc are combined is attracting much attention. This clutch

(Hele-Shaw) is fully described by its inventor in the Transactions of the Institute of Mechanical Engineers (Great Britain), July 1903. Fig. 38 shows a set, or "pack," of discs from such a clutch. Careful scrutiny reveals a V shaped circular impression struck up in the end disc.

127 Fig. 42 also shows the V shape of the discs very well indeed; in fact, the whole clutch is well shown here in section. I call attention to the female cone, *D*, bearing on the male cone, *f*, when the clutch

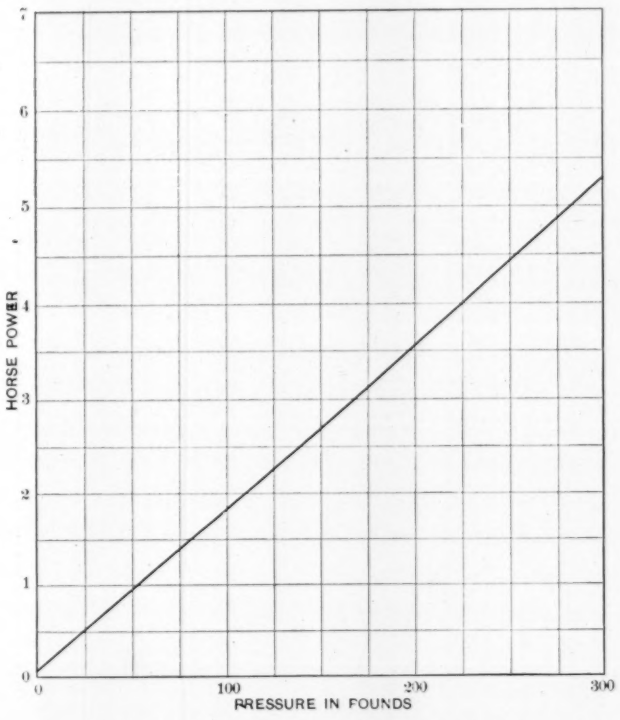

FIG. 43 POWER CHART FOR HELE-SHAW CLUTCH

is thrown out, thereby checking the spinning tendency of this clutch, or, if the viscosity of the oil is heavy, holding it quiet during the changing of gears.

128 In place of the entire surface of the discs bearing, only the V portions engage. This clutch is copiously lubricated and the V or engaging portions of the discs are perforated with holes so that the oil may circulate quickly in and out of the V grooves as they are engaged and disengaged. Outside the V portions of these plates or

discs there is a comparatively large space between them, permitting the free circulation of oil and consequent rapid carrying away of heat if the clutch slips much.

129 In connection with the article referred to in the Transactions of the Institute of Mechanical Engineers there are some very good data on power transmitted by various spring pressures, given in Fig. 43. Fig. 44 shows the character of the curve depending upon horse power and pressure of springs.

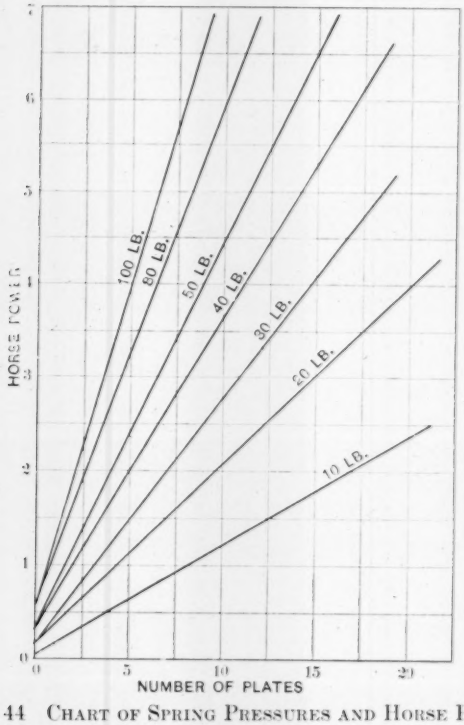

FIG. 44 CHART OF SPRING PRESSURES AND HORSE POWER

130 One-thousand horse power is being transmitted by one of these clutches running at 700 or 800 r.p.m. and measuring 18 in. in diameter between the V's in the discs. The following table gives the dimensions and number of plates used for different horse powers:

	Bronze	Steel
25 h.p., 27 plates of 6½ in.	14 outer	13 inner
40 h.p., 25 plates of 8½ in.	13 outer	12 inner
60 h.p., 21 plates of 11 in.	11 outer	10 inner

The space in length required inside of the clutch casing for 25 plates is 5 in., this including the space for the disengaging movement and the spring pressure plate.

131 The number of plates in this clutch is made to vary with the power transmitted, the diameters remaining the same within certain limits. The principle involved is that the thickness of the pack of plates shall not exceed the diameter of the plate. When this becomes necessary in order to transmit a load, the plates are increased in diameter, fewer of them being used. The clutch is necessarily heavy, but this is partially offset by the relatively small diameter. It has, consequently, little spinning tendency.

132 The materials for disc clutches in general have been various; namely, steel on steel, steel on leather faced discs, steel on bronze and steel discs with cork inserts.

133 I have recently been informed of a disc clutch with cork inserts of natural cork that wore out in about 1000 miles twice in succession. This same clutch was equipped with compressed cork inserts previously described, which have driven the car some 5000 or 6000 miles without perceptible wear.

134 It is a fact that steel discs against steel have become badly heated and cut to such an extent as to make the clutches inoperative. Steel against bronze, however, does not seem to cut in this manner and the wear after two years' steady use is only 0.002 in. or 0.003 in. at the outside edge of the discs. I have not heard of the original combination of Weston, that is wood against iron or steel, being used in connection with automobiles.

135 The cone clutch stands alone in the great care necessary to so construct it as to permit it to seat itself absolutely concentrically. All the other types of clutches are for the most part free from this difficulty. But it will be seen from the foregoing that the simplicity of the cone fully offsets the extra care necessary in the hanging or assembling of it.

136 It may seem from what I have said in regard to clutches in general that it is about the worst part of an automobile that can be mentioned, but I hasten to correct this impression if it exists. As a matter of fact, I find reference to the behavior of the clutches in the last Glidden tour, a tremendously severe test of some 1500 miles. An observer states that the clutches came out of this test quite as well as many parts of the running gear.

137 My own experience is that it is a mighty poor automobile clutch that cannot be neglected and cannot run without any attention whatsoever for 1000 miles.

138 A pneumatic clutch has been developed which has not been widely used because of its cost. It is a plain leather-faced disc pressing a metal plate as indicated in Fig. 45. The clutch is located within the flywheel and the air is forced from the pump through a small air cushioning tank and from there it enters the clutch through the hollow crank shaft *B* and an air valve. The air deflects the leather diaphragm *C*, causing it to bear against the metal disc *D* which can have a slight endwise motion, and forces it against the fiber disc *E* permanently riveted to the casing *F*, which latter is bolted to the flywheel.

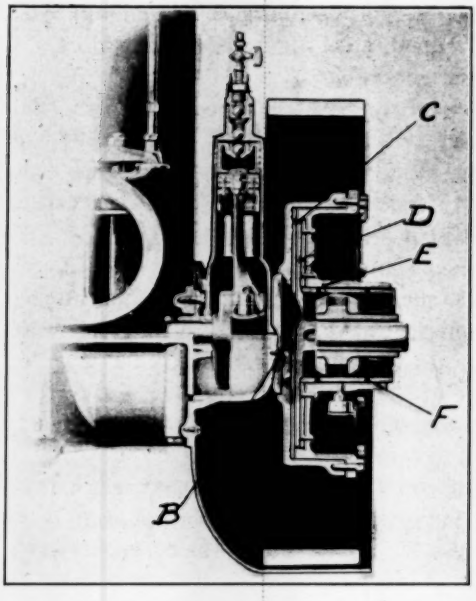

FIG. 45 AIR OPERATED CLUTCH

139 The makers, the Northern Motor Car Company, Detroit, give the following in regard to the clutch and its air pump:

The lubrication of the pump became a problem because we did not want the oil thrown into the air clutch, so an opening was left from the base of the pump into the upper part of the crank case, and this air seemed to be sufficiently oily to lubricate the walls of the pump without causing trouble in the air valves. This construction is extremely simple and cheap as far as the air clutch itself is concerned, but as a whole the air clutch is expensive, from the fact that it is necessary to build a very fine air pump and air valves, and conduct the air through

tight joints in the piping, and also drill out the crank shaft throughout its entire length in order to admit the air into the clutch, all of which costs much more than a good cone clutch.

140 Hydraulic clutches have been used but are not popular. Fig. 46 and 47 show one of the simpler forms.

141 The magnetic clutch is in use and is fairly successful. Such clutches are operated on the same principle as the so-called "pick up magnet" found in so many plants. One complication arises in the fact that one of the parts of the magnet has to rotate continuously, the gears being always in mesh; consequently the exciting current

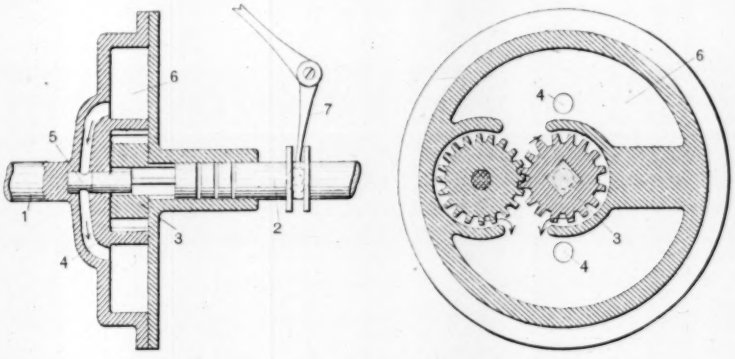

FIG. 46 FIG. 47

HYDRAULIC CLUTCH

has to be carried to it by a brush. These clutches do not heat badly, at least not badly enough to cause any trouble; but they seize rather savagely unless carefully controlled. A considerable current is also necessary on the car for the operation of the clutch. These complications rather interfere with extended usage.

142 As I have already stated, there is a great variation of detail in friction clutches. This paper cannot cover it all, but will, I trust, be of enough interest to lead to discussion, which will do more to fully develop the state of the art than has been possible in this summary which I have prepared.

DISCUSSION

DUTY TEST ON GAS POWER PLANT

By J. R. Bibbins and G. I. Alden, Published in Mid-November Proceedings

Mr. W. H. Morse I have been rather interested in the statement of Mr. Straub as to the rate of combustion per square foot as compared with his test. This test ranged from about 12½ to 17 lb., and if I am not misinformed, the horse power rating of this producer was about the same as that of the engine. I should like to know whether there were any indications during this test that the producers were operated close to their maximum rating.

2 How close to its rating (designed efficiency) did the producer operate? I have no information on this, excepting from the owners of the Norton plant, and I feel quite safe in saying that we could by no means have carried a full consumption rate, as the difficulties in carrying the heats of the fires would, in my judgment, have been insurmountable especially in continuing the test for 151 hours, which corresponds to a full week's practical run.

The Author Professor Reeve inquires why the producer efficiency curve is not flatter at full load. The characteristic curve of producer efficiency involves, as explained in Par. 26, an assumption that the losses in the producer bear a certain relation to the weight of coal fed into it; i.e., to the heat input. Our assumption, however, covers a comparatively short range of load, 400 to 550 b.h.p., so that no great error was involved in considering the two lines representing producer and engine input as parallel. Owing to the difficulty of running a sufficient number of coal consumption tests, we have no actual knowledge of the characteristic curve of producer efficiency.

2 Supplementary note: Since the original calculations upon producer efficiency were made, a new method of obtaining a fairly accurate knowledge of the producer efficiency at fractional loads has occurred to me. This is outlined below. In the average pro-

ducer plant, it is usually a difficult matter to conduct tests of the necessary duration at fractional loads. The only alternative that seems to meet the necessities of the case is to determine the average coal consumption during several periods in which the plant is kept running idle; and again, through standby periods, as, for instance, during Sunday in a plant not operating continuously. With a number of such runs averaged, fairly accurate data representing producer input at fractional loads, would be available. These may be applied as follows:

3 Referring to the accompanying curves, Fig. 1, the plot is identical with that of Fig. 8 in the paper, with the exception that the area

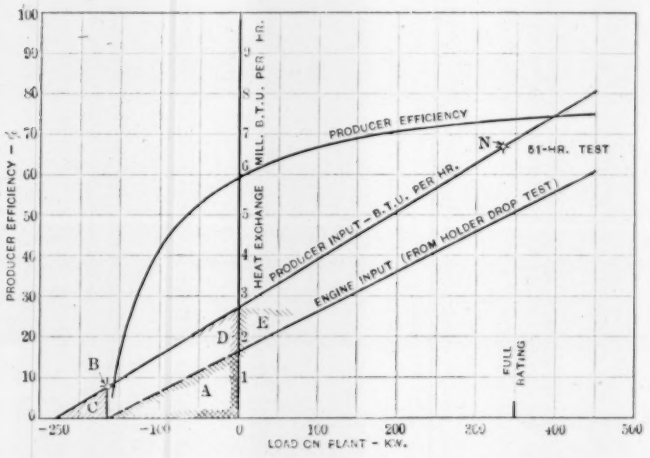

FIG. 1 METHOD OF ESTIMATING FRACTIONAL PRODUCER EFFICIENCIES

at the left of the origin has been included. The line of engine input is re-plotted from Fig. 2. As it is based upon a number of tests at different loads, we may assume that a proportional relation exists between load and heat input to the engine. Producing this line to the left, indicates that the losses (area *A*) inherent to the engine-driven unit, are 166 kw., covering mechanical friction, windage and all thermal and electrical losses up to the point of useful power generation.

4 But even with the engine standing idle, a certain loss is incurred in keeping the producer fuel bed up to the gasing point. Locate above this secondary origin a point *B* representing the heat value of

this standby coal.[1] Locate also a point N representing the heat input during the 51-hr. coal consumption test. We now have two points, B and N, with which to locate the line of producer input throughout the entire range of load. Producing this line to the left, indicates that the standby loss (area C) is equivalent to 70 kw., which is less than one-half of the no-load engine loss.

5 Total area D, or conversely, the rectangular area E at the right of the primary origin, now represents the total loss between coal pile and switchboard, which remains constant at all loads and may be considered as an overhead, or fixed charge upon the plant. Without this loss, it follows that the producer plant is capable of delivering power at the rate of 0.8 lb. per kw-hr. net at any load within its range of capacity; or in other words, its efficiency is constant.

6 From these two lines, the producer and engine input respectively, the producer efficiency curve may be obtained. At full-load, the efficiency is approximately 75 per cent, at half-load, 70 per cent, at no-load, 60 per cent—something of a paradox, as one naturally expects efficiencies to fall off progressively from maximum at full-load to zero at no-load. But it is interesting to note here that at no-load on the plant, the producer operated at a rate of practically one-third its normal output at full-load.

7 This efficiency curve is slightly flatter than that assumed in the paper, as embodied in Fig. 8; but, on the other hand, it seems to bear out in general the assumption made. The fact that the producer and engine input lines are not exactly parallel, indicates, first, a certain constant loss at all loads combined with, second, a variable loss increasing with the load, due perhaps to the increased loss of sensible heat in the gas, etc.

8 This cursory study indicates that the gas producer throughout the range of operation in the average plant is not only an unusually efficient piece of apparatus, but its efficiency is little affected by varying loadings. How best to conduct tests on this type of apparatus, is a matter for future consideration; but it is fortunate that the standby loss, both with the engine running and idle can be determined

[1]At a 500 h.p. producer gas plant at Richmond, Va., this standby coal consumption averaged 46 lb. per hour. At the Norton plant, the standby loss is somewhat higher owing to the extra loss incurred in rebuilding fires weekly. On the whole, however, the two different types of producers agree extraordinarily in this respect.

very simply, only one observation being involved, viz: the weight of coal, and an average of several runs should give accurate results.

9 Professor Reeve predicts a drooping efficiency curve, were the producer pushed hard enough. This seems to be a rather fictitious case, inasmuch as the operative difficulties surrounding any attempt to run at higher rates of gasification would prevent this point being determined—at least, such would have been the experience at Norton unless special equipment were provided.

10 Regarding the effect of temperature on gas readings: Unquestionably the temperature variation of the gas is much more important than that of the holder shell as effected by sun heat. Nevertheless, to avoid one possible source of error, the tests were run at night. As all measurements of gas quantity were made at the holder, a careful record of the pressure and temperature was made by the apparatus shown in Fig. 11 (paper), located at the holder outlet. By lifting the thermometer through the central tube provided, allowing gas to escape through the tube with considerable velocity, an accurate measure of the gas temperature was obtained. This varied but a few degrees throughout the test.

11 The frequency of water gas making for the entire test is shown by the gas log, Fig. 4, the time of each water gas run being plotted in per cent of the corresponding interval of air gas run. These runs were maintained fairly constant, and the respective periods seem to be dependent largely upon the condition of the fuel bed and the character of gas desired. With too long runs on water gas, the fires would be killed and steam would pass through without disassociation; with too short runs, the high temperature of the fuel bed would bring about excessive clinker formation. Some plants, requiring the maximum make of water gas, blast the fires for a shorter period, but at a much more rapid rate, thus keeping down the average temperature while increasing the output.

12 Mr. W. D. Ennis refers to efficiency tests conducted at Richmond upon a producer gas power plant of the same size and character, except equipped with a different type of producer. The principal reason for the somewhat lower efficiency shown is that the Richmond test was run at a considerably lower load than the average for the Norton test, even though above the generator rating. Consequently the curves, 1 and 2 in Fig. 1, accompanying his discussion, are not strictly comparable. Owing to the wide difference of opinion as to desirable over-load capacity of gas engine, the generators in this particular plant were much smaller than the corresponding load capacity of the

gas engines for continuous running. The Richmond test curve No. 2 should more properly be compared with Fig. 13 in the Norton paper, representing a large number of weekly operating results averaged graphically. These show about 1.8 lb. net coal per kw-hr. at a load corresponding to the Richmond test, which result checks fairly well with the latter when it is considered that the Norton plant runs but little over 30 per cent of the time with the 14-hr. standby losses included in these economies. By plotting the three tests at Richmond for various loads, the coal consumption of the plant at full engine load may be obtained by exterpolation, 1.59 lb. per kw-hr., equivalent to a little under 1.0 lb. per brake horse power hour.

13 Professor Kent deprecates the comparison of gas plant with what we may term the average steam plant. He cites some very high steam efficiencies and inquires how our efficiencies at Norton could have been improved. Perhaps we have made the comparison too abrupt, but I may again mention the fact that the Norton plant was tested in its regular everyday working condition, and as such, is comparable with the average operating results from a high grade Corliss plant. The general practice in steam engineering to base efficiencies on indicated horse power makes it more difficult to appreciate offhand the greatly improved efficiencies obtained from a good gas engine in which power is usually measured by brake or electrical horse power.

14 The Norton test results might have been improved in many ways entirely aside from the special care usually put upon an equipment to insure that it may be perfectly attuned to the requirements of the test. A steady rheostat load might have been substituted for the variable shop load occurring during the day, yielding a higher average load than recorded. The temperature of the jacket water might have been raised considerably, thereby decreasing the jacket absorption by a considerable percentage, part of which would be realized in increased engine efficiency. The producer fires might have been worked over continually to maintain more uniform conditions and better gas. Although the difference is not material, the engine will respond to an improvement in quality of gas, its capacity increasing with the heat value of the mixture. Take for instance the case of natural gas versus blast furnace gas, a large difference exists—perhaps as much as 30 per cent—in the heat value of a cubic foot of respective mixtures. Although the higher compression of the lean gas compensates to some extent, the latter evidently necessitates a larger cylinder for the same capacity, and hence a somewhat higher engine friction. And this is entirely independent of the question of combustion

within the engine cylinder. Any considerable variation in gas quality requires an adjustment of mixture at the engine. Hence, it is an unquestionable fact that to produce the best efficiency of combustion, the quality of gas must not be allowed to vary as it did to some extent during the Norton test (from 101 to 126 B.t.u. per cubic foot). It is, therefore, quite reasonable to predict an even higher plant efficiency than here recorded with the proper conditions maintained.

15 I appreciate the sympathy extended by Dr. Lucke in his comments upon the difficulties surrounding the testing of a producer of the intermittent type, and I am free to confess my total inability to analyze further, with the data at hand, the curious phenomenon of carbon deposition, which apparently took place within the producer. To my knowledge such a producer reaction has not been reported before, at least I have never noted any comments upon it from other investigators.

16 Mr. R. E. Mathot's method of computing mechanical efficiency does not seem to me applicable to a test of this nature. In fact, the efficiencies based upon the difference between full load and no load power, as constant loss would involve a very serious error in that the same conditions as regards either friction or combustion efficiency, do not obtain at full load. With increasing load, the engine friction unquestionably increases, due to the heavier bearing and pin pressures and thrust of piston rings against the cylinder walls. The former has been demonstrated by tests of large steam engines (e.g., 6500 h.p. vertical, three-cylinder Corliss, New York Edison, Waterside Station) and it should be equally true in the gas engine. The accuracy of the graphical method, in Fig. 5 of the paper, entirely avoids this assumption of constant engine friction, whether right or wrong. And its accuracy is proved by the almost perfect agreement of the two 10 hr. average points covering periods of fairly constant load which were plotted after the average line was drawn. This method is practically equivalent to calibrating the generating unit.

17 As to the accuracy of the indicator for gas engine work, I would emphasize the fact that with a slow-speed engine, such as we are dealing with, most of the difficulties of high speed work are avoided, and with reasonable care in the calibration of springs and in keeping the indicator clean and well lubricated, the most serious errors at least are overcome. From a detailed inspection of these 72 sets of cards, we feel reasonably sure of an accurate result; especially when the erratic results may so easily be detected, as by the graphical method.

18 As to the highest temperature that can be permitted for discharge jacket water inquired into by Mr. Parker, the average during the test held close to 110 deg. fahr. Individual water circuits, however, may have run higher, although we did not attempt to observe other than the total discharge from the engine. This represents normal practice at this plant, but the jackets might have run much higher, probably 150 deg. fahr. Some gas engine plants are operating on even hotter jacket water—one discharging at about 200 deg. fahr. in connection with a cooling pond; but 150 deg. may be considered a reasonable limit.

19 In reply to what Mr. Morse has said as to the probable maximum rate of combustion with the Norton equipment, I feel quite safe in saying that we could not have carried a much higher rate than the maximum during the test, 17 lb., unless this could have been brought about by operating the producer differently, and no experiments were made along this line. Certainly double the combustion rate would have been out of the question, considering the conditions obtaining after the 58 hr. continuous run, which corresponds to over a full week's commercial run.

20 Mr. Straub raises the vexed question of higher versus lower heat value of gases. This is an involved subject and one upon which a wide difference of opinion seems to exist. However, the analogy he introduces concerning the use of higher or lower heat value in fuel determinations, is, to my mind, hardly effective in influencing the argument one way or the other for the heat value of gases. At best, the difference is small as compared with gases in which there may be from 10 to 15 per cent difference between higher and lower values. The subject seems to me too complex to discuss adequately in this connection, and I can best refer to a communication from Mr. Arthur J. Frith in the discussion of the standardization of engine tests, Transactions, vol. 24. His discussion is brief and masterly, and I wish to add only one thought. Assuming the gas engine to be a non-condensible vapor engine, it is reasonable to stipulate that it should be supplied with a gas suitable for its use. Should, therefore, the engine be held responsible for wide and sudden fluctuations in the hydrogen content of gases? Is its efficiency as a heat transforming mechanism subject to sudden change, whenever the hydrogen content changes from one cause or another? Mr. Frith even raises the question of the application of the term "efficiency" to fuel gas; e.g., 100 per cent efficiency for a gas containing no hydrogen; and the efficiency decreasing as the hydrogen content increases.

THE SPECIFIC HEAT OF SUPERHEATED STEAM

By Prof. C. C. Thomas, Published in December Proceedings

Dr. S. A. Moss In Par. 3 of his discussion, Mr. J. A. Moyer makes some remarks on calculations of specific heat of superheated steam from laws of impulse force of jets of superheated steam.

2 The writer has done a great deal of work in this direction, and has found that conclusions such as those drawn by Mr. Moyer are not valid. The impulse force of a jet of superheated steam, while directly dependent upon specific heat, changes very little for different values of specific heat. Therefore, a law for impulse force, such as Mr. Moyer alludes to, while it may be *almost* exactly correct, will nevertheless give erroneous values of specific heat.

3 Laws of impulse derived from observation are, of course, subject to slight errors and hence can never give reputable values of specific heat. In other words, impulse force is such a slowly changing function of specific heat that it is not legitimate to use values of the function for exact determination of specific heat.

4 Computations in the inverse way; that is, calculation of impulse force laws from values of specific heat, show that wide differences in specific heat give insignificant changes (within the limit of observational errors) in impulse force laws. Hence the conclusions regarding specific heat which Mr. Moyer draws from impulse laws cannot be depended upon.

The Author The following statements are given in answer to some questions in the foregoing discussion.

2 The values given by the curves of total heat of superheated steam include the total heat of saturated steam as found in the ordinary steam tables based upon Regnault's determinations. It is highly desirable that new determinations of heat of vaporization should be made, in order to test the accuracy of existing data.

3 In measuring steam temperatures, the cold end of the junction was kept in melting ice, in a Dewar bulb, the other end was introduced directly into the steam, as shown in Fig. 3, at E.

4 Fig. 5, 6, 7 and 8 indicate that the specific heat becomes practically constant at zero pressure, as suggested by Dr. Moss, excepting near the saturation point.

5 Fig. 9 is to be regarded as representing the culminating and final results of these experiments. A study of Fig. 10 to 15 will show that

the experiments there represented were made before those represented in Fig. 9, and while less regular than the latter, are corroborative in all respects of the final results in Fig. 9. By the time the experiments recorded in Fig. 9 were made the apparatus had been perfected to such an extent and such skill in operation had been attained, that the curves given in Fig. 9 could be experimentally reproduced at will, and the same results obtained time after time. The introduction of the independently heated air jacket as a heat insulation about the calorimeter proved to be the last step required in order to obtain the regular results shown in Fig. 9.

6 In all specific heat of steam experiments with electrically heated calorimeters, of which the writer has knowledge, with the exception of those which yielded the results given in this paper, the question as to the external work done by the steam in expanding through a calorimeter has required consideration. This was true of the first experiments made by the writer, but the difficulty of dealing satisfactorily with this question led to the method finally adopted, of using only one thermometer, in a fixed position, upon which both initial and final temperatures were measured. The method of doing this is described in Par. 21 to 28.

7 The slightest change in the quality of the steam entering the calorimeter is detected with ease, as described in the paper. A constant weight of steam is passing per unit of time, and being dried and then superheated to a given temperature by a constant electrical input of energy. If the amount of evaporation required varies in the slightest degree the temperature to which superheating takes place changes with the change in quality. The question of initial quality of the steam is thus effectually disposed of.

8 The method of making computations from the data given in Table 1 is fully set forth in Par. 59. These computations have been made with the greatest care, have been repeatedly checked and the results plot into the smooth curves shown in Fig. 9.

9 A careful reading of the paper will show the relation between the curves in Fig. 10 to 14, and Fig. 9, and the value of Fig. 10 to 14, in corroborating the curves in Fig. 9.

10 In certain of the illustrations, especially in Fig. 10 to 12, imperfections in method of reproduction of the curves have caused a number of dots and blotches to appear like experimentally determined points. The real points in these figures are all on or quite near the curves, and it will be noticed that in Fig. 9 the true points are almost absolutely upon the curves.

11 The writer trusts that this closure will answer the most important of the questions which have been asked concerning the paper.

COLLEGE AND APPRENTICE TRAINING

By Prof. J. P. Jackson, Published in October Proceedings

The Author In closing the discussion, I desire to correct a few wrong interpretations, that were taken from my paper.

2 Prof. D. C. Jackson said I had "an axe to grind," and I had. My suggestions are exactly in line with his argument, that we should have more coöperation between the industries and engineering schools. The paper deals with one practical phase of such coöperation: the joining of both these interests in the making of a technical, or business, engineer.

3 Mr. Hammerschlag says, in Par. 5, "For those who have completed a technical course * * * ask the industries to take them on their face value." This is just what I ask. The technical graduate is not, and likely never will be, immediately upon graduation, ready to step into a position of responsibility; therefore, let the industries which alone are able, by reason of their equipment, give the needed additional preparation. Mr. Hammerschlag says that my paper neglects the individual. I am unable to understand where. He also says that the special apprentice, or graduate student engineers' course "raises class and cast in the shop," makes "men and foremen resentful," "is narrow," "is demoralizing * * * like a man that is being nursed," pays less than "for digging trenches," "destroys * * * loyalty." Are these not theoretical statements based on conventional premises? Such conditions should not and need not exist; nor do they, where I have made an examination. Mr. Hammerschlag's argument against such courses is much weakened by the statement that immediately follows: "the Carnegie Technical Schools have had most cordial coöperation and assistance by the industries of Pittsburg, and my remarks were not made with respect to coöperation in that district." Does he mean that graduates of this school only get generous, broad treatment by the Pittsburg industries? Or is Pittsburg the only district in which technical graduates get such treatment?

4 Mr. Porter's discussion is largely based upon a supposed error in my statement and a misconception of the purpose of the graduate student engineers' courses. If he will inquire more seriously (Par. 7), of the various companies named in my paper, he will find established courses, as I represented; and if he goes further in his investigation,

he will find other courses of a similar character, both established and in course of development. Thus, I must insist that my premises are correct, as the evidence in my hands is incontrovertible.

5 His second mistake is his thought that the object of my paper was to show how the industries can "secure their skilled workmen" (Par. 2). After this statement, practically all his arguments, other than those intended to show that my facts are wrong, are based on the assumption that the apprenticeship courses dealt with, are for the training of mechanics, rather than engineers, and certainly not for college men. This is fully evidenced by the courses (Par. 14 and others); and the trade schools (Par. 15 and 16) referred to by him.

6 The "fundamental principles," or instructions to colleges, named by Mr. Porter in (Par. 18 and 20 to 27), have not been unthought of by students of applied pedagogy, and those of utility have been in more or less general use for some time.

7 Mr. Rice is a man who has evidently gone through such a course as I described; and, as a result, is thoroughly competent to speak upon the subject. In the main, he upholds my contentions; and is my best witness to the usefulness and success of the educational movement under consideration. Mr. Tompkins' excellent discussion upon the shop training of workmen deals largely with the apprenticeship systems for other than college men and, therefore, does not directly enter into this discussion.

8 Professor Ennis and Mr. Taylor apparently have a clear conception of the problem as it appeals to me. Most of the suggestions they make are of a nature to aid in the development of technical courses for graduates such as will tend most surely to turn out in the end good engineers or staff officers. There seems no good reason why in such courses the man cannot be taught the meaning of a "hard day's work," learn to read the thoughts and "feelings of the workmen" in the shop, and be brought against suitably "keen competition" (Mr. Taylor, Par. 5, 6 and 7).

9 Mr. Jones shows clearly the gap between the end of the college course and the position of responsibility in the industries. His suggestion that this gap may possibly be filled by modifying the college courses cannot, in my estimation, be answered affirmatively. It may be lessened, but some real contact with an industrial organization will always be required before the graduate is prepared for his true function. If this contact can be obtained in the shops before graduation, excellent; but why not, at least in part, after?

10 It has been stated in this discussion that, if the workmen are

suitably trained, there will be evolved from their ranks plenty of good officers. I absolutely disagree with that statement, however much I feel the urgent need of industrial education for workmen and I aver that much of the rapid advance and improvement, throughout the whole range of industries of this country and Europe, is in large measure the result of collegiate technical education. It might as well be suggested that the Army and Navy give up West Point, Annapolis, and their various practical officers' training schools with the belief that, if the rank and file are excellent, all needed officers would automatically develop.

11 Permit me to reiterate that my paper deals with the subject of *making officers for the industries,* be they line or staff. I incline to believe that college men appreciate thoroughly the need of preparing men for the *business* or *staff side* of the industries, as well as the technical, and are modifying the college courses with that object in view. The purpose of the paper must not be confounded with the magnificent movement now on foot in this country for the proper *training of skilled workmen,* both by industrial schools and apprenticeship systems. The student engineers' courses, now in existence, can and will, undoubtedly, be greatly improved upon; but they are now performing, however crudely, a necessary function. As a proof of their success, I find that large numbers of men who have taken such courses have, almost uniformly, quickly risen to positions of responsible usefulness. My plea is that these practical training schools for officers are of great utility; and deserve careful attention from the managers of industrial corporations; further, that their proper development requires a scientific, practical knowledge of both shop management and pedagogy.

INDUSTRIAL EDUCATION

By W. B. Russell, Published in October Proceedings

The Author The statement has been made that the plan described is applicable only to special cases and to rich and powerful corporations. The methods of shop instruction would appear to be quite general in their application. Mr. Gantt has called attention to the value of the shop instructor for training men as well as apprentices. A number of railroad officials have expressed the opinion that all departments of the railroad should be recruited by similar methods. Of the school training it may be said that all industrial education is special in the sense that the illustrations and problems are selected in

each case from the shop or industrial environment. This does not in the least interfere with making the training general and even to some extent cultural in its results.

2 As to the scheme being applicable only to rich and powerful organizations, the small shop has the advantage in dealing with apprentices. There is then no occasion for a central organization like that required by a large railroad. In the small shop there is the added advantage of closer personal contact between employer and employed, and often there is less specialization. Some of the most successful solutions of the apprentice problem are at present being worked out in small manufacturing plants and small railroads, and results are being obtained which it will be hard to equal in large corporations. It is expected that the system described will ultimately be extended to shops having five or more apprentices.

3 The point is well taken that there is danger of too much emphasis on mechanical drawing and of too little upon problems demonstrating shop practice. It was necessary at first, however, to emphasize the drawing in order to demonstrate at once to the shop authorities the value of the school work, and this plan has been entirely successful in producing immediate benefits. More attention is now being given to other subjects and additional equipment has been procured for problems and experiments. In apprentice courses, the danger to be guarded against, is not that of over education, but of misdirected education, as there cannot be too much training of the right kind.

4 The discussions by Professor Lanza and Professor Williston should remove any misapprehension in the minds of educators or others who imagine that apprentice training is intended to rival that of technical schools. Apprentice education is in a field by itself and is at present supplementary to other types of instruction.

5 The movement has been characterized repeatedly as philanthropic. While it may be true that the first idea of the originator of the plan was one of philanthropy, and while it is true that all of the instructors do much more than that for which they are paid, it is nevertheless a fact that the movement is a plain business proposition and would cease to exist if it were not. The reason that the plan looks like philanthropy is because of the economic fact that everything which tends to give an apprentice added skill, breadth of view or development of character makes him that much more valuable to his employer. That which is best for the boy is best for the company. It is thus possible for a broad-minded, far-sighted shop management to develop a boy in the lines for which he is best adapted and to the

full limit of his ability, and still to do this as a business proposition. Training the sons of employees in such a way is a common point of agreement between men and management.

6 Professor Lanza has hit upon the keynote of the class work when he says "the education given to any class of men should be the best for that class." Professor Williston calls attention to the fact that the railroad is dealing, not with a picked body of boys, but with the rank and file. The railroad is giving that which the public school has failed to give, an education by which a boy can earn a living.

7 In spite of the obstacles mentioned, the *esprit de corps* is not so different from that in other schools. Rivalry between schools is keen and is fostered by visits of instructors and boys to other shops of the system. Apprentice clubs are the order of the day, with ball teams and inter-shop games.

8 Mr. Keep says that what is needed is an industrial educational system devoid of specialization. The author would agree with this in so far as it means that the State should do much that the manufacturer and the railroad are now forced to do. However, when the State does take hold of such training it will be difficult to find a plan founded on educational principles more broad than the plan described, in which the methods and much of the subject matter is of universal application and is so regarded by the best educators. It may be of interest in this connection that one city of the Middle West is adopting the same methods and portions of the apprentice drawing, problem and physics courses for its public school system.

9 It is well to realize that whatever may be the ideals of apprentice training, any plan is doomed to failure that does not take into account actual commercial and industrial conditions. While there is a field for all types of industrial education it remains a fact that comprehensive apprentice training is destined to play no small part in the solution of our present industrial problem.

THE FOUNDRY DEPARTMENT AND THE DEPARTMENT OF ENGINEERING DESIGN

By W. A. Bole, Published in September Proceedings

The Author Regarding the use of fillets, I do not object to the use of any except those which are too large. It would be equally as bad to omit a fillet where it is required as to make it excessively large and have the interior bleed away. A fillet of some size is necessary at practically every junction of two walls of a casting, and my point is that a small fillet is often far better than a big one.

POWER SERVICE IN THE FOUNDRY

By A. D. Williams, Jr., Published in October Proceedings

The Author Mr. Richards has called my attention to the fact that main compressed air lines can be made tight and kept tight; but there are many cases where they are not kept tight. A large leakage occurs in nearly every compressed air system in the connections made to the various driven machines, particularly when these connections are of a temporary nature. Undoubtedly leakage in a compressed air line and in all of the machine connections can be practically prevented, but it is very rarely done.

2 I should have qualified my statement that "compressed air lines, as usually constructed, are designed to remain tight only long enough to pass the acceptance test." I did not have the kind of acceptance test in mind which is usually associated with this expression, but the rough and ready test made in the field by the pipe fitter, or the erecting man. Such tests are much more frequent than those of a purely scientific nature.

3 Mr. Richards and I evidently have different ideas as to what constitutes a small compressor. Compressors dealing with less than 60 to 70 cu. ft. of free air per minute do not have Corliss valve gears.

4 In the class of service to which Mr. Richards has alluded, subaqueous tunnel work, reliability in a compressor carries far greater weight than steam or power economy. A saving of five to ten pounds per hour in steam consumption during several years would not make up for the damage resulting from one enforced shut down of the air compressor lasting any considerable time.

5 Mr. Mumford's remarks I agree with, in all particulars. Mr. Ronceray's remarks are interesting and his observations have been confirmed in my own experience.

6 In regard to the relative economy of compressed air or electric transmission: Mr. Johnson doubts the economy of such transmissions for less than five miles. In this the individual case has so much to do with the cost of construction that it would be easy to assume conditions or to find conditions where electric transmission would be the more economical, or the reverse. Aside from economy of operation and construction there are many cases in which a small electric driven air compressor, merely by its convenience and the ease with which it can be carted around and got into operation, would be far ahead of the pipe line system of getting the air to the work.

7 Mr. Lane has spoken of the locomotive crane. This machine

should be better known in the foundry, and as it can be utilized as a shifting engine in addition to the numerous ways Mr. Lane has suggested, it merits some attention. The locomotive crane renders the foundry man independent of the delays of railroad switching service around his own plant. My reply to Mr. Richards covers the points in regard to compressed air lines mentioned by Mr. Lane.

FOUNDRY BLOWER PRACTICE

By Walter B. Snow, Published in Mid-October Proceedings

The Author It was not the purpose of this paper to present the actual results of specific tests; relative performances only are shown as indicative of average conditions and results. Air was not assumed as an incompressible fluid, but allowance is made in the formula for its change of density relatively to the pressure. The significant figures are given because so calculated by the given formula. It is merely a matter of opinion whether round numbers should be substituted. The formula employed and the results given are those generally accepted by blower manufacturers.

DESIGN OF ENGINES FOR THE USE OF HIGHLY SUPERHEATED STEAM

By Max E. R. Toltz, Published in September Proceedings

The Author The writer is pleased that this subject has aroused so great an interest, as shown by the discussion. In answer to several questions he can state that the tables regarding the economy of superheated steam were compiled from actual tests by Professor Hrabak.

2 The remarks regarding coal performance of locomotives made by Mr. Emerson are rather surprising. There is no doubt that better results can be obtained under test than in actual practice, but it should be borne in mind that the conditions differ in both cases. In practice a locomotive may be out on a run for from 15 to 20 hours, while its actual, continuous work is only 8 to 12 hours. The balance of the time it is standing on side tracks, etc., which means that coal must be spent for non-productive work. But there is not one case to the knowledge of the writer in which there has been shown such difference in coal consumption, viz: 80 lb. against 240 lb. per 1000 ton miles.

3 Superheaters on locomotives in foreign countries have been a success during the last five years and in this country and Canada very fine results have been obtained. The Canadian Pacific has over 350 engines equipped, while in the United States only 22 are running, 7 of which the writer designed. The average saving of coal over the saturated steam locomotive on the Canadian Pacific is from 15 to 18 per cent, while on the Great Northern, 20 per cent has been recorded by the monthly performance sheet, yet in tests 28.4 per cent have been obtained.

4 The cost of manufacture of poppet valves and their maintenance has been fully answered by Mr. Ode.

5 Referring to lubrication, the oil which has given perfect satisfaction in stationary engines is Gargoyle-Hecla B, with a flashing point of 650 deg. fahr., manufactured by the Standard Oil Company, while on locomotives a similar product of the Galena Oil Company is being used. Although forced feed is recommended the latest types of hydrostatic lubricators have done the work satisfactorily. The superheat on locomotives runs from 175 deg. fahr. to 250 deg. fahr., with a final temperature of the steam of from 550 deg. fahr. to 625 deg. fahr., without developing any difficulty in lubrication. It is generally assumed that a greater quantity of oil is required for lubricating valves and cylinders where superheated steam is used, but experience seems to show just the reverse, although positive statements to this effect cannot as yet be made.

6 The statement made by one of the discussers that with 60 deg. superheat at the throttle of an engine 18 deg. were still recorded in the exhaust steam with a vacuum of 24 in., cannot be correct. Take for instance a 20 in. by 30 in. simple engine, working with a 30 per cent cut-off and a steam pressure of 160 lb., it will be necessary to superheat the steam 165 deg. in order to have the exhaust steam 212 deg.

7 In regard to increase of boiler pressure, attention is called to the fact that we are aiming to decrease same, especially in railroad practice. Of course, in that case it will be necessary to increase the size of the cylinder.

8 In answer to Mr. Seymour's remark in regard to decreasing the efficiency and reliability of a plant as a whole, there is no difficulty in operating properly designed engines with superheated steam, because engines can be simplified for this purpose. In other words, a simple engine using superheated steam will have a higher economy than a compound engine using saturated steam and a compound engine using

superheated steam will give better economy than a triple expansion engine working with saturated steam. The heat consumption drops rapidly with superheating up to the point where the superheat is sufficient to produce saturation at the end of expansion, and it still decreases when superheating is carried still higher. It therefore should not be concluded, as in the example given by Mr. Seymour, that the economy with highly superheated steam is but little greater than with moderately superheated steam. If in the example given the speaker had mentioned the different details as to steam pressure, cut-off, type of engine, etc., the writer could probably have replied to this point more intelligently.

9 The lubrication of the cylinder for highly superheated steam is an easy matter when it is properly designed and the piston supported at both ends of the cylinder by liberal bearings.

10 The writer did not intend to convey the idea that the poppet valve is the only proper valve for engines using superheated steam but it is better adapted for that purpose than the Corliss valve because the latter has larger wearing surfaces and cannot therefore be satisfactorily lubricated, especially when steam is used having a temperature of over 500 deg. fahr. Attention is called to the fact that with four piston valves attending to the steam distribution, very good results have been obtained.

11 The writer perfectly agrees with Mr. Seymour that a just comparison of the efficiency of engines can only be drawn from the heat consumption per horse power hour, but in the final reckoning the economy of any steam plant must be figured at the coal pile. The writer has endeavored to show this in the three tables submitted, but at the same time he calls attention again to the diagram 1A, in which the British thermal units per horse power hour for steam of different temperatures are given. In regard to the extra expenditure of British thermal units in superheated steam over that in saturated steam, the following example will give an idea of the economy: A pound of saturated steam having a pressure of 100 lb. absolute, contains 1181.2 B.t.u. and has a cubical content of 4.34 cu. ft., which means that each cubic foot of this steam consists of 272.35 B.t.u. If this steam is to be superheated 200 deg. fahr., 96.1 B.t.u. have to be added, which increases the total British thermal units of this pound of steam to 1278.2, while the volume has been enlarged to 5.5 cu. ft. Therefore, each cubic foot of this steam contains now only 235.55 B.t.u. or 13½ per cent less than the cubic foot of saturated steam of the same pressure. The heat added amounted to 8.13 per cent, but the increase

in volume is practically 26 per cent. In a 20 in. by 26 in. slide valve engine the port has a cubical content of 660 cu. in., or 0.4 cu. ft. At a cut-off of 30 per cent the superheated steam will give a saving of 7 per cent in clearance alone, which is due to the increase of the volume.

12 In regard to the difficulties of superheating and the troubles and tribulations in using superheated steam, the writer has had opportunities to investigate many cases and he will endeavor to point them out and show the remedies which can be applied. For instance, mention was made of a big power plant equipped with separately fired superheaters. Tests made by Professor Hrabak show plainly that the economy with such superheaters is not very high. A separately fired superheater in which the gases go to waste has no place in any power plant except where it can be fed with waste gases, such as are plentiful in steel or iron works. By proper attention such a superheater can maintain a fairly uniform temperature in the steam and this is probably the reason why this type seems to be in favor in this country.

13 Where the superheater is located in the gas passages and is part of the boiler, the economy in steam and especially in coal will be marked, if the proper degree of superheat is applied and uniformly maintained. These two factors, the establishment of the proper degree of superheat and the uniformity of same throughout the different loads a boiler is subjected to, are very important.

14 It is generally conceded that a Corliss valve engine will operate successfully with steam at a temperature of 475 to 500 deg. fahr., but in one instance which came under the observation of the writer recently a temperature of 525 deg. fahr. was reached without impairing the working of the different parts of the engine. Under such conditions a high flashing oil must be applied for lubrication at the right time and place.

15 Taking it for granted that a temperature of 500 deg. fahr. is allowable and assuming the steam pressure to be 165 lb. with a temperature of 373 deg. fahr., to be raised 127 deg. by superheating, the per cent gain in steam and coal economy in different types of engines should be as follows at normal load:

	SIMPLE NON-CONDENSING		COMPOUND CONDENSING		TRIPLE EXPANSION CONDENSING	
	Direct	Indirect	Direct	Indirect	Direct	Indirect
Steam	21	21	15	15	10.5	10.5
Coal	8.5	16	1.5	10	−3	6.5

16 But there is not a single power plant in this country which shows such results, although in Europe they are obtained in every day practice. What is the reason for not deriving the same benefit here as in Europe? First of all, European engines are designed for high superheat and can stand the stresses of expansion and contraction due to great fluctuations. Second, the degree of superheat is controlled or regulated according to conditions. Taking, for instance, the above case, the heating surface of the superheater to give 127 deg. fahr. superheat is proportioned for normal load, but any increase of load will naturally increase the degree of superheat, so that with an overload of 50 per cent the superheat will be about 25 deg. higher and with 100 per cent overload about 60 deg. higher. Trouble ensues in consequence and in all cases the superheater heating surface is cut down so as to give at maximum load the degree of superheat wanted, while at normal load the superheat is about high enough to give dry steam.

17 Another factor of irregular and variable superheat is to be found in the elements of the superheater which in most cases consist of round pipes in which only the steam nearest to the wall is superheated, while the core is hardly heated on account of the outer ring of superheated steam acting as a non-conductor. In some superheaters devices are arranged to break up this segregation and mix the steam of different temperatures thoroughly, but generally this is not done. In consequence, the steam reaches the engine in alternate gusts of different degrees of heat and plays havoc with valves, pistons and cylinders, which, of course, discredits at once the advantages of superheated steam. The elements of a superheater should be so constructed that the steam flows through them in a sheet in which form a uniform superheat can be obtained.

18 Another failure in a superheater is the depositing of cinders, soot and ashes on the outside of the elements, which cuts down the efficiency. In many cases flooded superheaters are used, but if anybody wishes to increase his troubles, he needs only to employ this type. Another source of trouble lies in the connection between headers and elements being exposed to the hot gases, which generally ends in leakage. It is, therefore, recommended that the headers and their connections with the elements be located outside of the boiler, which guards against such leakage.

19 The control or regulation of superheat has been accomplished in different ways, but there are only a few which have merits.

a Injecting cold water into the superheater elements. This

is not recommended because in time the superheater pipes will be filled up with the incrusting elements of the water.

b By-passing the flue gases or choking the flue passages by dampers. Not rational, because in the latter case the flue area is restricted when it should be largest.

c Admitting cold air onto superheater elements. This is wasteful.

d Superheaters reinforced by great masses of cast iron to store up heat. Costly and also ineffective to a certain degree.

e Hot water circulation through pipes located inside of superheater elements. Too complicated.

f Mixing superheated steam with saturated steam automatically in a comingler.

g Returning excess heat of superheated steam into boiler, thereby not alone increasing evaporating capacity, but improving water circulation.

These last two systems are rational and economical and can be automatically arranged.

20 Last but not least, the degree of superheat to be used is very important. Superheating of steam, although not new, is a peculiar feature in steam engineering and it is not as yet fully understood by all. There are many factors that must be taken into consideration before it can be determined what is the most economical amount of superheat in a given case. What should be the design of engines or turbines for new or old plants and what can be done to improve present conditions as they are found? What are the probable losses in pipe lines? What is the proper type of superheater for a particular plant and what should be its location? Many recent installations have been failures because such points as these have not received attention. These are all engineering problems that should not be left to the manufacturers of engines and boilers, or of other steam appliances, but should be worked out by an expert who is advanced in the art. I venture to say that in the best modern and most economical high duty power plant in this country, using saturated steam, the economy can be raised from 8 to 10 per cent by a judicious application of superheated steam and that the superheating apparatus will not give any more trouble in such a plant than a first-class boiler.

21 The turbine builders have made the statement that with superheated steam there is economy in steam consumption in turbines, but

that no economy in coal consumption has been recorded, and for that reason they recommend only a moderate superheat of about 100 deg. fahr. This is entirely misleading and should be corrected. The tests quoted were on a plant having a separately-fired superheater, the construction of which was of the crudest form, and the fuel arrangment a very wasteful one.

22 In a paper read before the American Street and Inter-urban Railway Association, Mr. A. H. Kruesi gave the results of tests at Chicago with a Curtis turbine coupled to a 9000 kw. generator. The lowest superheated steam consumption of the turbine when developing 13 650 h.p. is given at 9.62 lb. of steam per horse power hour, which probably is the lowest steam consumption of any in this country.

23 Comparing this with results obtained with a 3000 kw. Curtis turbine, built by the Allegemeine Electricitats Gesellschaft of Berlin, at an output of 2554 kw., the steam consumption per horse power hour was 8.3 lb. and at an output of 3169 kw., 8.47 lb. per horse power hour. The turbine ran at 1500 r.p.m., the boiler pressure was 170 lb. and the superheat maintained 220 deg. fahr. This turbine was built especially for highly superheated steam and special care was taken to provide for expansion and contraction of the different parts, and, as the writer understands it, the openings as well as the angles of the different sets of blades were somewhat differently designed to give the steam the proper expansion due to superheat.

24 The writer has tried to explain the features of superheating and their difficulties as thoroughly as his knowledge enables him to do and he can say that in the present state of the art, it is not a question of whether we shall superheat or not, but whether low or high superheat shall be applied. With low superheat very few changes in engine design are necessary, but with high superheat more attention will have to be paid to rational details as outlined in the writer's paper resulting in a high degree of economy which to aim at is worth our while.

CONTROL OF INTERNAL COMBUSTION IN GAS ENGINES

By Prof. C. E. Lucke, Published in Mid-November Proceedings

The Author From the discussions that have been presented regarding some of the indicator card waves it is evident that I have not made myself clear in the body of the paper on the differences between explosive waves and indicator inertia waves. The

explosive wave, as is indicated by its other name of "detonating wave," is generally a manifestation of violent momentarily localized pressure which would start oscillations of the indicator mechanism. These indicator oscillations are easily distinguishable by their periods and damping characteristics and the fact that they are present in most of the cards I have presented does not prove the contention of some of the gentlemen who have spoken on the subject that the explosive wave is absent. It rather proves that the explosive wave was of a detonating sort, which may or may not have persisted as long as the indicator wave. It must be borne in mind that there is ample evidence to prove that the indicator wave would not be present with a properly selected spring unless the detonating or explosive wave had first appeared. It is extremely likely that the gas wave does not persist as long as the indicator wave, and this fact is probably the reason for many misinterpretations.

FOUNDRY CUPOLA AND IRON MIXTURES

By W. J. Keep, Published in November Proceedings

The Author In answer to the questions that have been asked I would say this: So far as the walls of the cupola are concerned, most of the cupolas in the market are straight and very much alike. I simply presented this construction as one that has done good work. I know that those who used the straight wall cupola get just as good results. I will change the melting ratio in my paper from 18 to 10 to 1.

2 Answering Professor Bird's remarks in regard to the scrap, I agree with him if it could be done. If a piece of scrap is about the size of the casting, and of the grain that you expect the casting to have, remelting the scrap will close the grain, partly on account of the sulphur that is in it, and partly on account of the low heat used in remelting.

3 There seems to be a great variety of opinions among foundrymen as to the practical use of borings. It is a matter of fact, however, that all who have endeavored to close the grain of iron and prevent sponginess have found the use of cast iron borings successful. It was a patent process by Mr. Whitney, the car wheel manufacturer, but the patent has expired and it is now public property. It seems to be one of the suggestions that always helps the foundryman out of trouble.

4 In regard to the recovery of the iron from the borings, they should be treated in the way which Mr. Whitney suggested. Pack them in boxes holding 100 lb., nailing the cover on tight, and then charge them just the same as 100 lb. of iron. It is obvious that the boxes will reach the melting point without being burned, and when they reach the free oxygen the wood will burn off and the borings will come out in small quantities and there will be very little loss. I presume about 10 per cent would be the maximum loss and sometimes it would be less.

5 Answering Mr. Smith's inquiry in regard to a one-inch bar: It will be difficult to reply to the question except in a general way. The strength per square inch of a test bar is in proportion to its size, that is, to its rate of cooling. An inch square bar is stronger proportionally than one or two inches square; the exact relations between the strengths cannot be arrived at by the formula ordinarily used.

6 At the meeting in Chicago in 1904 I presented a table which is not only interesting, but is very useful, giving multipliers and divisors by which the strength of bars of different sizes can be calculated.

7 As to casting a test bar vertically or horizontally, my impression was that a test bar cast vertically was stronger than when cast flatwise. The experiments made by our former testing committee, however, showed the contrary to be true.

8 Several years ago I was asked by a Western university to make a number of test bars for them to determine the influence of aluminum which we added to our iron and I cast them vertically, supposing it would give them more strength, and then in order to have the test bars look well, I put them in a tumbling mill and made them smooth. The result was that those test bars proved to be so strong that I never heard the result of the tests. I found out afterward, however, by Mr. Outerbridge's experiments, that the tumbling of cast bars, or the tumbling of any other casting, increases the strength of the material perhaps 25 per cent.

A FOUNDRY FOR BENCH WORK

By W. J. Keep and Emmet Dwyer, Published in Mid-October Proceedings

The Authors The 12 or 14 ft. that we have under the floor is ample room for molding machines and it seemed to be better to put them there. We want a carrier that will not occupy too much room in height and that will take a flask from the molding machine and shoot

it clear to the other end of the floor and bring a mold back and shake it out at the machine.

2 I might say that we did not intend to manage everything automatically as Mr. Mumford seems to infer.

THE STEAM PATH OF THE TURBINE

By Dr. C. P. Steinmetz, Published in March Proceedings

Prof. Sidney A. Reeve I feel somewhat backward at offering any discussion on this paper, as I saw it yesterday for the first time and have read it only about one-quarter way through. I find it has aroused in my mind, as in Mr. Longwell's, certain questions with regard to the mathematics embodied in the fundamental equations of the paper. I think Mr. Longwell is right in saying that the interest of the paper centers in the earlier portion of it.

2 As to the equation for the energy I should like merely to state that I corroborate Mr. Longwell's view; that there are two expressions for the available energy. The one used here is for the external work available from an isolated quantity of gas, in which the translational energy only is available. But in the use of steam, whether in reciprocating engine or turbine, we have another portion of energy available. We cannot get the quantity of steam into our steam engine or turbine, without having boiler-expansion taking place. The work available during expansion comes not only from the translational energy of the particles of the steam, but also from the rotational and disgregative energy of each molecule, which develops through the expansion; so that the equation of the perfect gas does not apply. If to Dr. Steinmetz's expression for the available energy of an expanding gas, or $\int_{p_2}^{p_1} p\,dV$, which is true within proper sense, the energy available from the steam's entrance to the turbine, or $\int_{o}^{V_1} p_1\,dV$, and then subtract the negative energy of exit of the steam against back-pressure, or $\int_{o}^{V_2} p_2\,dV$, the algebraic sum of the three (all of which fulfil Dr. Steinmetz's insistence that the work done is the total force times the increment in displacement) will result in $\int_{p_2}^{p_1} V\,dp$, which is Mr. Longwell's expression and is the correct one.

3 On p. 276 I have noted that it might help the casual reader somewhat if it was noted that pressure stated in kilograms per square

meter, divided by 10 000, is closely equal to one atmosphere. That line on p. 277 is approximately figured in atmospheres.

4 I also take issue with Dr. Steinmetz where he brings in his superheat as a negative of moisture. Of course, that is one way of expressing superheat mathematically, and a way in which it may be done if done correctly. But Dr. Steinmetz's expression of the superheat as a negative moisture has been derived from volume ratios. If he carries that same argument into the field of superheat he must remember that his gamma and delta express volume ratios. But immediately below (in equation [24]) he jumps from the volume basis to an energy basis; which is not permissible, of course, unless volumes and energies are always mutually proportional, which is not true except in the moisture field. So I should take issue instantly with him as soon as he expresses his minus gamma (equal to delta) in terms of energy, instead of in terms of volume.

5 On p. 279 he determines the maximum velocity attainable in the jet by assuming the final pressure to be equal to zero. We know that the pressure cannot go to zero unless all other natural quantities become either zeros or infinities. The velocity must then be the square root of infinity or infinity also. It is inconceivable that we can have any finite velocity determined by a pressure range from any initial pressure to zero back pressure. In fact, there is no such thing in nature as zero pressure.

Mr. H. E. Longwell I trust that you will appreciate the feeling of trepidation with which I attempt to take issue with so redoubtable a mathematician as Dr. Steinmetz. There be those with seven-league boots who can leap lightly from mountain top to mountain top in the domain of pure mathematics, but most of us have to keep close to the ground, and reach our destination through long and tortuous paths so little traveled since our college days, that their unfamiliarity gives us a constant feeling of uncertainty.

2 The paper is essentially a mathematical one, and as such the only point open to discussion is whether the basic formulae represent the actual facts. With the basic formulae established, mathematics is either right or wrong. There is no room for difference of opinion here.

3 To my mind the interest centers in the first half of the paper. The special consideration of the steam turbine is fairly elementary in its character, and I do not think there are any theories advanced that are not pretty generally accepted.

4 The development of the equations for the pressure, volume,

moisture and energy of saturated steam, however, constitutes an exquisitely beautiful mathematical study, and I hope I do not display ignorance in saying that it has the charm of novelty.

5 The principal question I raise is: Does Dr. Steinmetz's basic formula for energy, as given in Par. 5,

$$E = \int_{p_1}^{p_2} p \, d \, V$$

represent the physical fact?

6 Turning to Par. 17, it will be noted that the numerical values of E and v given therein for the several pressure limits, as determined by Dr. Steinmetz's formulae, do not accord with the calculations made by the common entropy-temperature method. By this method, expanding from 180 lb. to 14.7 lb., E is 43 000 mkg. instead of 39 000 and v is 918 m. instead of 875 m.

7 From pressure 14.7 lb. to 28 in. vacuum, E is 40 300 mkg. instead of 36 000, and v is 889 m. instead of 840.

8 From 180 lb. to 28 in. vacuum, E is 78 220 mkg. instead of 75 000 and v is 1238 m. instead of 1210.

9 In this paragraph we have the obvious absurdity of the energy of saturated steam expanding from 180 lb. to 28 in. vacuum, being identical with the energy of saturated steam expanding from 180 lb. to 14.7 lb. plus the energy of saturated steam expanding from 14.7 lb. to 28 in. vacuum.

10 It is quite evident that the values for E and v between the limits of 180 lb. and 28 in. vacuum have not been calculated from the formulae but have been assumed to be equal to the sum of the energies and the square root of the sum of the squares of the velocities calculated for expansion between 180 lb. and 14.7 lb., and between 14.7 lb. and 28 in. vacuum, respectively, the steam being initially saturated in such instance.

11 This will account for the smaller discrepancy between the numerical values given, and those calculated by the entropy-temperature method, in the case of the larger ratio of expansion.

12 I have not verified any of the numerical values given for E and v except for expansion from 180 lb. to 14.7 lb. These agree substantially with the formulae, but as they differ considerably from the values obtained by the older methods, we are led to suspect the truth of the formulae.

13 This is where integration comes in. The only kind of integral that is of any use to you is one that you know by sight. If you know

it only by reputation it can't serve you very well. If you will allow me, I will draw you pictures of two that I am quite well acquainted with personally.

14 In Fig. 1 and 2 let the curves AB represent the relation between the pressure and volume of saturated steam expanding adiabatically. The horizontally shaded area in Fig. 1 is a speaking likeness of

$$\int_{p_2}^{p_1} V\,d\,p$$

and represents to the best of our knowledge the energy of saturated steam expanding from p_1 to p_2.

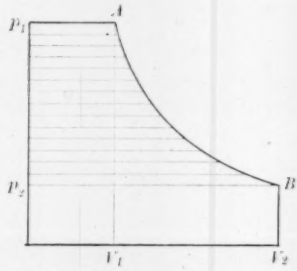

FIG. 1

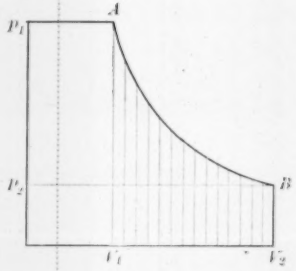

FIG. 2

15 Likewise the vertically shaded area in Fig. 2 would be recognized anywhere as

$$\int_{V_1}^{V_2} p\,d\,V$$

and I don't know what it represents. The formula for energy given in Par. 5 is

$$\int_{p_1}^{p_2} p\,d\,V$$

16 We seem to be able to go through all the motions of integrating $p\,d\,V$ between the limits p_1 and p_2, but it is an unusual operation, and I am unable to form any conception of what the integral looks like or what it represents. At any rate it does not seem possible that it represents the energy of saturated steam expanding adiabatically, as determined by generally accepted methods.

17 If we take the more rational formula for E, that is to say,

$$\int_{p_2}^{p_1} V\,d\,p$$

we develop a very satisfactory energy equation by methods considerably less intricate than those adopted in the paper.

18 Substituting equation [12],

$$V = \frac{S}{p^{1-c}\, p_o^{c-i}}$$

we have

$$E = \frac{S}{p_o^{c-i}} \int_{p_2}^{p_1} \frac{dp}{p^{1-c}}$$

$$= \frac{S}{p_o^{c-i}} \left\{ \frac{p_1^c - p_2^c}{c} \right\}$$

$$= \frac{S\, p_o^i}{c} \left\{ \left(\frac{p_1}{p_o}\right)^c - \left(\frac{p_2}{p_o}\right)^c \right\}$$

or for saturated steam, when $p_1 = p_o$,

$$E = \frac{S\, p_o^i}{c} \left\{ 1 - \left(\frac{p_2}{p_o}\right)^c \right\}$$

19 This expression is identical with Dr. Steinmetz's equation [14]

$$E = \frac{S\, p_o^i}{a - 1} \left\{ 1 - \left(\frac{p_2}{p_o}\right)^c \right\}$$

except as regards the denominator of the fractional coefficient.

20 The value of a being assumed as 1.126, the value for E is $1\frac{1}{8}$ times that obtained from Dr. Steinmetz's equation, and conforms very closely to the value obtained by the entropy-temperature calculation. Unless I have made a grave error, the paper will require revision as regards all of the subsequent equations based on this energy equation [14].

21 I have not had time to review thoroughly any more of the fundamental equations, and perhaps this is as much of a purely mathematical discussion as you would wish to listen to.

THE AUTHOR My reason for preferring the exponential form of steam equations is their superiority for determining the general relations between quantities entering into thermodynamic problems. While the practical calculation of a steam engine or turbine can be carried out by the use of steam tables, a study of the conditions of economy and efficiency leading to an advance in the art, requires the

expression of the relations between the variables by a general equation; and even such approximate equations are superior to tables of numerical values. The usual steam equations are not as well suited for general investigation as the set of equations which I give in the first part of my paper.

2 The exactness of a numerical calculation is not the exactness with which values can be computed from the tables, but is the exactness of the data from which the tables are calculated.

3 A difference of 10 per cent between values calculated by two different sets of approximate equations obviously is not reasonable. The explanation of the difference mentioned in the discussion is found in the accidental omission of the factor a, resulting from the terms admission and exhaust, in equations [13] to [17], and under the square root of [28] to [30]. Thus corrected, the numerical values in Par. 17 of the paper read:

$p_1 = p_2 =$	127 000	$p_2 =$	10 350	$E =$	43 600	$v =$	925
$p_1 = p_0 =$	10 350	$p_2 =$	690	$E =$	40 100	$v =$	887
$p_1 = p_0 =$	127 000	$p_2 =$	690	$E =$	80 300	$v =$	1240
$p_1 = p_0 =$	127 000	$p_2 =$	0	$E =$	178 000	$v_0 =$	1870

4 The beginning of Par. 5 of the paper should read:

"The work done by the adiabatic expansion from pressure p_1 to pressure p_2, or the available energy of adiabatic expansion, is:

$$E_1^2 = \int_{p_1}^{p_2} p \, d \, V$$

and the total steam energy between pressures p_1 and p_2 is:

$$E = \int_{p_1}^{p_2} p \, d \, V + p_1 V_1 - p_2 V_2$$

and the numerical constants in [13] and [15] should be changed from 78 700 to 88 000.

5 The value $a = 1.126$ of the adiabatic constant of steam agrees with the experimental data within the errors of their observation, so that no conclusive experimental evidence exists on the variation of this constant with the moisture of the steam. Theoretically however, it can be shown that this constant a must decrease at very high percentages of moisture, as when the moisture approaches 100 per cent a must drop below e. Within the range of moisture appearing in engines and turbines, the variation of a can be neglected.

6 Steam adiabatically expanded to zero pressure would have a finite velocity, since an infinite velocity would lead to the irrational assumption that a finite quantity of steam contains an infinite amount of energy. While the absolute zero of pressure can no more be reached than the absolute zero of temperature, both terms are physically justified. The graphical meaning of the finite values of velocity and so of energy of adiabatic expansion to zero pressure and so zero volume is, that the area between the $p - V$ curve and the V axis as asymptote, from any finite value of V to $V = \infty$, is finite.

7 In the approximate equation of the specific heat of superheated steam [21], the temperature has not been introduced. As the values of c_p given in the literature were very unsatisfactory, a careful investigation was made on this subject, which showed the c_p to be independent of the temperature, at least within the range which comes into consideration here. However, equation [21] can be considered only as an approximate expression of the observed data.

8 Regarding the representation of the superheat as negative moisture: As superheat increases, moisture decreases the potential energy of steam, the former can be represented numerically by the negative of the latter, or inversely, irrespective of whether the steam follows the same equations in the range of superheat as below saturation, or not; and in either case this representation is justified.

9 As the constants of superheated steam are very incompletely known, and their numerical values questionable; as the amount of additional energy resulting from superheat is small compared with the total energy; and as our extensive investigations of nozzle reactions and steam jet velocities show that in passing from superheat to moist steam no marked break occurs in the steam curves, it appears permissible to use the same equation until more information on the constants of superheated steam is available, and the constants and exponents in equation [24] then can be modified accordingly.

10 Regarding the efficiency of steam expanding in a turbine nozzle: As this quantity is of fundamental importance in steam turbine design, a very extensive set of investigations was made during several years, covering the entire range of pressures and pressure ratios, superheat and moisture, and nozzle throat and mouth sections met with in steam turbine work. It was found that at the pressure ratio corresponding to the nozzle section, as discussed in section 3 of my paper, the expansion in the nozzle is almost perfectly adiabatic. That is, the mean velocity of the jet issuing from the nozzle (as meas-

ured by weighing the reaction of the nozzle) reaches within 2 per cent or 3 per cent of the theoretical, and the velocity in the center of the jet (as measured by the impact tube) coincides with the theoretical velocity of adiabatic expansion. Some such nozzle velocity curves are given in Fig. 1 herewith, with the ratio of the mean jet velocity to the theoretical velocity as ordinates, and the nozzle mouth as abscissae.

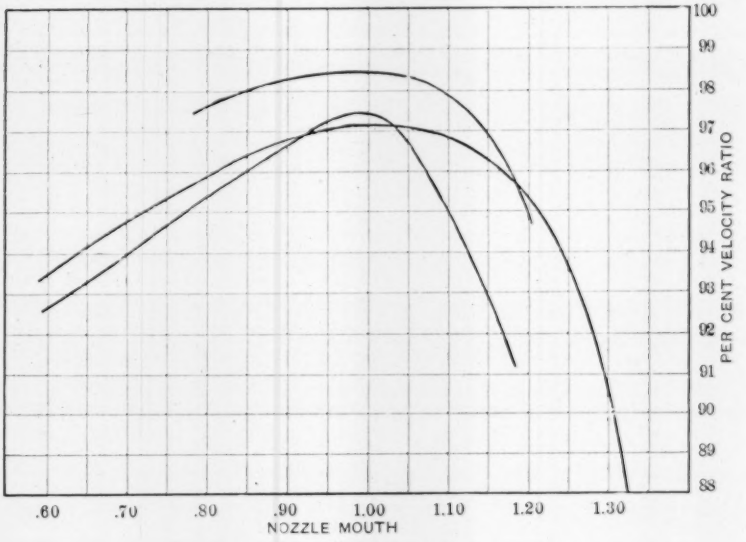

FIG. 1 NOZZLE VELOCITY CURVES

11 The experiments illustrated by Mr. S. L. Kneass are very interesting, but hardly have any bearing on the subject; because:

a Only his last nozzle could be expected to give reasonably efficient expansion.

b A thermometer inserted in the steam jet does not show the temperature of the jet, but the temperature produced by the conversion of the jet velocity into heat due to impact with the thermometer, and its reading therefore is meaningless.

c The transparency of the steam jet at the nozzle mouth does not disprove the presence of moisture, as moisture particles become visible only after conglomerating to a size comparable with the wave lengths of visible light.

12 Some other points mentioned are of minor importance or have been withdrawn since the discussion. The reason for using the metric system I have given in the footnote. Perhaps, also, having started in the field of electrical engineering where the metric system of units has always been used exclusively, the English system appeared to me more objectionable than to those who are not familiar with the simpler system of units. In the English system, some of the equations in cubic feet, pounds per square inch, and foot pounds, are:

$$p_o V_o = 326\, p_o^{\,i}$$

$$E = 426\,000\, p_o^{\,i} \left\{ \left(\frac{p_1}{p_o}\right)^c - \left(\frac{p_2}{p_o}\right)^c \right\}$$

$$v = 8 \sqrt{E}$$

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ENGINEERING REMINISCENCES. Contributed to Power and American Machinist. By Charles T. Porter, Honorary Member A.S.M.E., author of "A treatise on the Richards Steam Engine Indicator and the Development and Application of Force in the Steam Engine, Mechanics and Faith. *John Wiley & Sons, New York.* 8vo, cloth, 335 p., 52 illustrations, and 38 full-page portraits. Price, $3 net.

Contents by chapter headings: Birth, Parentage and Education; in the Practice of Law. Introduction to Centrifugal Force; Invention and Operation of a Stone-dressing Machine; the Evolution and Manufacture of Central Counterpoise Governor; Introduction to Mr. Richards; Invention and Application of my Marine Governor; Engineering Conditions in 1860; I meet Mr. Allen; Mr. Allen's Inventions; Analysis of the Allen Link; Invention of the Richards Indicator; My Purchase of the Patent; Plan my London Exhibition; Engine Design; Ship Engine Bed to London; and sail myself; Arrival in London; Conditions I found there; Preparations and Start; My London Exhibit, its Success, but what was the matter? Remarkable Sale of the Engine; Sale of Governors, Visit from Mr. Allen; Operations of the Engine Sold to Easton, Amos & Sons; Manufacture of the Indicator; Application on Locomotives; Designs of Horizontal Engine Beds; Engine Details; Presentation of the Indicator at the Newcastle Meeting of the British Association for the Advancement of Science; Contract with Ormerod, Grierson & Co; Engine for Evan Leigh, Sons & Co., Engine for the Oporto Exhibition; Getting Home from Portugal; Trouble with the Evan Leigh Engine; Gear Patterns from the Whitworth Works; First Order for a Governor; Introduction of the Governor into Cotton Mills; Invention of my Condenser; Failure of Ormerod, Grierson & Co.; Introduction to the Whitworth Works; Sketch of Mr. Whitworth; Experience in the Whitworth Works; Our Agreement which was never Executed; First Engine in England Transmitting Power by a Belt; The French Exposition of 1867; Final Break with Mr. Whitworth; Study of the Action of Reciprocating Parts; Important help from Mr. Frederick J. Slade; Paper before Institution of Mechanical Engineers; Appreciation of Zerah Colburn; The Steam Fire Engine in England; Preparations for Returning to America; Bright Prospects; Return to America; Disappointment; My Shop; The Colt Armory Engine Designed by Mr. Richards; Appearance of Mr. Goodfellow; My Surface Plate Work; Formation of a Company; Mr. Allen's Invention of his Boiler; Exhibition at the Fair of the American Institute in 1870; Demonstration to the Judges of Action of Reciprocating Parts; Explanation of this Action; Mr. Williams' Instrument for Exhibiting this Action; Boiler Tests in Exhibition of 1871; We lose Mr. Allen; Importance of having a Business Man as President; Devotion of Mr. Hope; Close of the Engine Manufacture in Harlem; My Occupation During a Three Years' Suspension; Production of an Original Surface Plate; Efforts to Resume the Manufacture; I Exhibit the Engine to Mr. Holley Contract with Mr. Phillips; Sale of Engine to Mr. Peters; Experience as Member of the Board of Judges at the Philadelphia Centennial Exhibition; Engine Building in Newark; Introduction to Harris Tabor, Engine for the Cambria Iron and Steel Company; My Downward Progress; My Last Connection with the Company; The Fall and Rise of the Southwark Foundry and Machine Company; Popular Appreciation of the High-speed Engine.

THE BLAST FURNACE AND THE MANUFACTURE OF PIG IRON. By Robert Forsythe. *David Williams Company. New York. 1908.* 8vo, 368 p., illustrated, cloth. Price, $3 net.

Contents by chapter headings: Introductory; Commercial Classification of Iron; Constitution of Pig Iron; Physical Properties of Cast Iron; Materials of Manufacture; Preparation of Ores; Fuel; Fluxes; Description of Plant; Operation of the Furnace; Burdening the Furnace; Action within the Furnace; Furnace Irregularities; Hints on Design and Equipment; Supplement; Appendix.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. *Transactions. Vol. 12, 1906.* 8vo, 376 p., cloth.

AMERICAN CERAMIC SOCIETY. *Transactions. Vol. 9, 1907.* 8vo, 808 p., paper.

ASSOCIATION OF ENGINEERING SOCIETIES. *Journal. Vol. 40, no. 2. February 1908.* 8vo, 127 p., paper.

BROOKLYN ENGINEERS CLUB. *Proceedings. Vol. 11, 1907.* 8vo, 178 p., cloth

ENGINEERS' CLUB OF PHILADELPHIA. *Proceedings. Vol. 15, no. 1. January 1908.* Paper.

AMERICAN GAS INSTITUTE. *Proceedings. Vol. 1 and 2, 1906, 1907.* Annual. Large 8vo, 1021 p., cloth.

INSTITUTION OF ELECTRICAL ENGINEERS. *Journal. Vol. 40, no. 187. February 1908.* 8vo, 234 p., Paper.

MERCHANTS ASSOCIATION OF NEW YORK. *Year Book. 1908.* 8vo, 91 p., paper.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS. *Transactions. Vol. 15, 1907.* 4to, 252 p., 114 plates, cloth.

WESTERN SOCIETY OF ENGINEERS. *Journal. Vol. 13, no. 1. -February 1908.* Paper.

NEW EXCHANGES

MADRID CIENTIFICO. *Vol. 15, no. 590. March 1908. Madrid.* 8vo. Price, 20 francs a year. Paper.

REVISTA MARITTIMA. Monthly. *March 1908. Rome. Vol. 41, no. 3.* 8vo, paper. Price, 25 francs a year.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

08 Partner wanted to start a general engineering business in Chicago. Must be familiar with power plant equipment and electric installations, able to invest one-half the cash to put the company on a working basis. Give experience and references.

09 Good blast furnace draftsman; considerable amount of designing, principally furnace stack, hoist and stock house. A man who is simply a draftsman will not answer; one who has had some experience in this line of work. Place would be an agreeable one for a man with ambition. Location, Virginia.

010 Rare opportunity for the right man. Fully equipped machine shop for general machinery repairs, and manufacturing, with an established business, for sale at a sacrifice owing to long continued illness of one of the proprietors. New York.

MEN AVAILABLE

71 Designer of special machinery, mechanical superintendent, or as plant engineer.

72 Member, graduate electrical engineering, Massachusetts Institute of Technology; five years experience, leading consulting and contracting engineering companies; eleven years large street railway company. Office and field designs, calculations, construction, tests and operation of steam electrical and hydraulic plants, apparatus and railway equipment.

73 Member, desires position as engineer in charge of construction and maintenance of power plant or mill work. Cement mills a specialty; also familiar with high pressure hydraulic work, steam and electric work in general, blast furnaces, level and transit, foundations, concrete, structural and mechanical details.

74 Member, Lehigh graduate, age 37. Ten years experience in management mechanical department coal mining property and railroad, with extensive shops.

Large experience hoisting and pumping machinery, air compressors and motors, steam locomotives and cars. Position similar to above, or as works manager or superintendent.

75 Wanted position as superintendent or works manager by a member, desiring a change. Wide experience, technical and practical, tool building, gas engines, producer and producer engines, steam pumps, special and automatic machinery and foundry practice. Twelve years shop practice. Ten years factory executive. Will contract to produce given results; can furnish A1 reference.

76 Member, located in Chicago, desires connection with first class firm or consulting engineer. Has wide experience in construction, design and operation of large industrial plants, valuation and buying. Accustomed to handling large quantities of business economically and rapidly.

77 Graduate, mechanical and electrical engineer. Experience in erecting light and power plants. Last three years commercial engineer and contractor in Spanish speaking country.

78 Manager or superintendent, age 33, American, married. Foundry or machine shops, plant equipment engineering; successful organizer of men, methods and principle; competent in manufacturing, sales, purchase, and acted as secretary. Applied technical training supplemented with broad experience in industrial engineering. A1 executive, strong personality.

79 Mechanical engineer, 20 years experience in the design of steam and compressed air engines and heavy transmission machinery, would prefer position as advisory engineer and estimator where theoretical and practical knowledge of steam engineering and of mechanics of materials as applied to machine design and general construction is especially desired.

80 Junior, mechanical engineer, technical graduate, seven years experience, design and construction, Prefers permanent position at moderate salary with good opportunity to advance.

81 Technical graduate, 35 years old, successful experience as superintendent, salesman and manager in this country, France and Germany, will be open for engagement May 1. Experience has been on light and medium interchangeable work.

82 Member, technical graduate, 16 years in charge of design of light and medium interchangeable machinery and the tools for its production, also plant construction and maintenance.

83 A designer with experience on special machinery, tools and fixtures for interchangeable manufacturing, including presses and dies for sheet metal working. Can handle help. Prefers location in the Chicago district.

84 Stevens, M. E., eleven years experience, including design and manufacture of elevating and conveying machinery, design of power plants and cement mills, is now open for engagement.

85 Mechanical engineer open for engagement; technical graduate; extended experience in steel and manufacturing plants as apprentice, machinist, boiler and engine man, draftsman, designer, fuel economy expert, engineer, chief and consulting engineer. Has executive ability and can get a maximum amount of work out of men under him. Experience in cutting down costs and increasing output; familiar with modern methods of cost keeping. Can furnish the best of references from former and present employers.

86 Position is desired as chief engineer or superintendent with firm engaged in or taking up the manufacture of gasolene trucks; associate member; four years continuous experience in this work, from draftsman to superintendent; technical education, executive ability, systematic, energetic. Available May 1; present salary, $2100.

87 M.E., Stevens, 1898. Ten years experience power plant construction, factory equipment and maintenance, thoroughly acquainted with up to date methods of factory management and accounting. Desires position as superintendent of maintenance, construction, or as mechanical engineer with contracting firm.

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